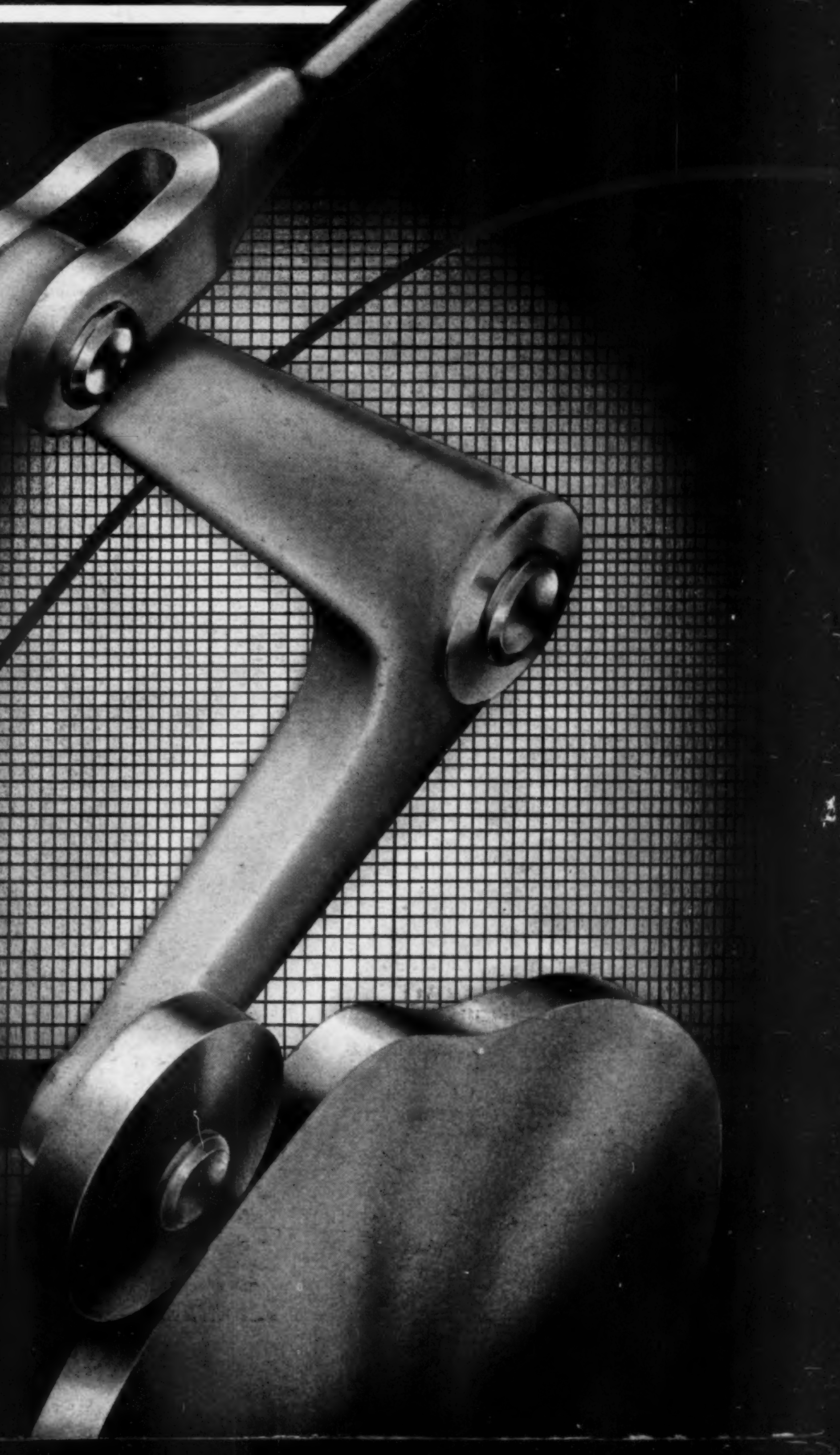
MACHINE DESIGN

April 1951



Machine Drives and Controls

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MACHINE DESIGN

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MACHINE DRIVES AND CONTROLS ISSUE

DESIGN FOR PRODUCTION • STYLING • MATERIALS SPECIFICATION • DESIGN ANALYSIS • MACHINE COMPONENTS • ENGINEERING MANAGEMENT

Over the Board

Return Post Cards

Hundreds of our readers are using the handy return post cards to request copies of articles for their files. As we expected, some articles are vastly more popular than others and we have even run out of stock on a few—notably "Systematic Design of Mechanisms" by Ziebolz (December) and "Stress Concentration Factors" by Peterson (February data sheet). Fortunately we are able to furnish copies of the Ziebolz article in reprint form. The Peterson data sheet has not been reprinted, but the material contained in this and the other four data sheets of the series will be included in a book to be published later by John Wiley & Sons Inc. Watch for the announcement.

Questionnaires

This is open season on readership questionnaires. A lot of our advertisers and their agencies have been making studies to find out what magazines are read by engineers. Our favorite magazine seems to be yours too, because we've seen several of these studies where MACHINE DESIGN led all the rest. Thanks a million for your loyal responses. We'll continue to do our best to justify your choice.

This Month's Cover

Symbolizing machine drives and controls, the cam and follower mechanism pictured in four colors on the front cover is a disembodied spirit. We expect the imagination of our readers to supply the input power and to fit the drive to a suitable task. The motion curve in the background should supply a clue as to its character. Execution of the design, which also appears as an introduction to the main editorial section on Page 121, is the work of Penton artist George Farnsworth.

Design Abstracts

Although most of the articles in MACHINE DESIGN are original—either contributed or staff written—many developments closely related to the field are reported for the first time to the various engineering societies such as ASME and SAE. Each month we study the papers presented before the major societies and present abstracts of those having possible interest to you. The lead-off abstract usually is relatively comprehensive (see Page 206); succeeding ones are briefed usually by concentration on only one or two aspects of the subject. Sometimes only the introduction and conclusions from the original paper are presented. In each case the aim is to give a quick picture of the subject—an outline of a technique useful in design, an engineering-interest story on a special development, or words of wisdom on the human side of engineering. Complete copies of the papers usually may be obtained by writing to the headquarters of the societies themselves.

Flying Shears

Need for cutting or shearing a moving web or section of stock has always presented designers with a real problem. With solid materials such as metals, the cutting blades are required to shear straight through and retract without breakage. Doing this at the speeds billets pass through the mill has proved to be difficult. The brief article "Electronic Shear Control" on Page 162 shows a neat design solution for this problem. There has also been found a need for cutting soft pliable materials which are extruded at rates which vary during the processing. To shear exact lengths of such stock the knives must detect the rate of instantaneous travel, adjust automatically and shear without upsetting the material. A solution for such a design problem should be interesting. Have you one?

Crossed Shafts

A recurring problem that seems to plague many of our readers is how to transmit power between nonparallel shafts which do not intersect. A fairly obvious solution is to use helical gears which do an acceptable job so long as the loading is light. Recently a reader informed us that he was having considerable difficulty with such a drive. Transmission of 50-hp peak loads resulted in a gear life so short as to be impractical. Despite considerable published information on the limitations of helical gears when used on nonparallel shafts the word seems not to have penetrated into all quarters. Is there a better practical solution?

Versatile Cells

Not many of us realize that metallic rectifiers are being effectively employed in control circuits as current limiting valves, nonlinear resistors and magnetic-amplifier components. In the short span of a decade, these rectifier cells have been developed to the point where their characteristics no longer change due to aging but are economical and reliable sources of direct current for precise controls. In the article "Selenium Rectifiers" beginning on Page 163, the author discusses many applications in which these versatile cells are employed including voltage regulation, dynamic braking, current blocking, magnetic amplifiers, and aircraft relays.

ITEMIZED INDEX

Classified for convenience when studying specific design problems

MACHINE DESIGN is indexed in Industrial Arts and Engineering Index Service, both available in libraries generally.

TOPICS

LEANER-ALLOY STEELS intended to stretch supplies of critical alloying elements have been announced by AISI. The new steels employ smaller quantities of nickel, chromium and molybdenum than comparable NE steels of World War II. Hardenability and toughness are retained by the addition of a noncritical alloy containing the element boron.

FREE-PISTON POWER PLANT developed in France combines the thermal efficiency of a conventional diesel engine with advantages of gas-turbine operation. A supercharged 2-cycle diesel, without the usual connecting-rod and crankshaft linkages, produces exhaust gas at 75 psi and 900-950 F. Power is generated by expansion of the gas to atmospheric pressure in a gas turbine.

PLASTICS as replacements for metals as an emergency measure is not optimistically viewed by the Bakelite Corp. Plastics most likely to replace metals in durable goods are phenolic laminates and a type of glass laminate employing styrene polyester. Bakelite reports that production of both is restricted by shortage of raw materials, which are being required increasingly in other areas of the defense program.

NEOPRENE can be applied by brush or spray gun as an air-dry protective coating on steel, concrete or wood. Produced by Gates Engineering Co., the new coating is claimed to be resistant to oil and grease, many chemicals, and age-cracking by sunlight and weather.

DRY ICE has found unique application in the rubber industry. It is used in tumbling operations to help remove the molding fins from rubber products. The dry ice freezes the fins, making them brittle, and the tumbling action breaks them away.

DIELECTRIC PROPERTIES of polyethylene and other plastics have been utilized by the Goodyear Tire & Rubber Co. in a new air filter. Plastic, as a porous mass of thinly shredded films, is the filtering element. When exposed to a current of air, the plastic mass picks up an electrostatic charge and attracts and retains fine dust, soot or smoke particles suspended in the air stream. When dirty, the filter can be cleaned and fully regenerated by immersion in water containing a detergent.

RADIOACTIVE ISOTOPES, as tracers, are helping to broaden knowledge about steel. For example, radiocarbon 14 is being used in studies of the distribution of carbon in steel, its segregation habits and the mechanism by which it causes steel to harden. Also, friction problems are being attacked in several ways that make use of isotopes. Still another use permits the thickness of pipe walls to be determined while the pipe is in service.

CHROMIZING PROCESS developed in England is reported to have bridged the gap between the electrodeposition of chromium and the alternative use of solid stainless steel for corrosion-resistant and decorative steel products. Chromium is diffused into the steel and chemically bonded and alloyed with it. Depth of the stainless zone can be varied normally from 0.004 to 0.010 inch.

SOLDERS with reduced tin content have been announced by the American Smelting and Refining Co. The addition of silver permits tin to be reduced by 50 per cent or more.

EXTRUSION-DIE POOL is being set up by USAF in co-operation with industry to eliminate bottlenecks in the aircraft and the aluminum and magnesium extrusion industries. This interchangeabiilty program is a step toward standardization of the many extrusion dies and specifications used in the manufacture of aircraft for the government.

MULTIPLE-THREAD HOBS increase production and have longer life, compared to single-thread hobs, according to a report by D. A. Moncrieff and H. Pelphrey of Michigan Tool Co. before a recent ASTE meeting. Increased life results particularly in climb hobbing because of better chip load distribution. Multiple-thread hobs should be larger in diameter than single-thread hobs to permit use of more flutes. Spiral gashing is preferred over straight.

MACHINE DRIVES and CONTROLS

SUPPLEMENT TO
MACHINE DESIGN
APRIL 1951

Machine Drives and Controls

IT HAS been said that the design of a machine is a perpetual series of decisions whether to stay in a rut or to try something new. No branch of engineering presents more opportunities—and hazards—than the design and application of machine drives and controls. Drives and controls are the brains, nerves and muscles of a machine—the things that distinguish it from a mere static structure. Herein lie the problems that really challenge the ingenuity and engineering skill of the designer.

Modern highly developed mechanisms—mechanical, electrical and electronic, hydraulic and pneumatic—confront today's designer with a choice that could be bewildering. Shall he stick with the tried-and-true or shall he endure the headaches of applying the radically new on the chance of spectacular improvement but with the possibility of failure?

Fortunately the headaches are not as bad as they used to be. The engineering principles that underlie mechanism dynamics, electrical phenomena and fluid flow are being constantly applied to the improvement of machine components. Diverse applications of the newer mechanisms are continually adding to our knowledge of their behavior. The specialized experience of any one engineer is no longer a reliable guide to design; in short, much individual experience quickly becomes obsolete.

To keep our readers up to date in this large field of machine drives and controls is one of the leading objectives of MACHINE DESIGN throughout every month of every year. Once a year, however, we focus attention sharply on the subject by devoting the major portion of the enlarged April issue to machine drives and controls. The following pages contain a comprehensive roundup of articles which it is hoped will stimulate thinking and provide working tools that can be used in current and future designs.

Colin Carmichael

EDITOR

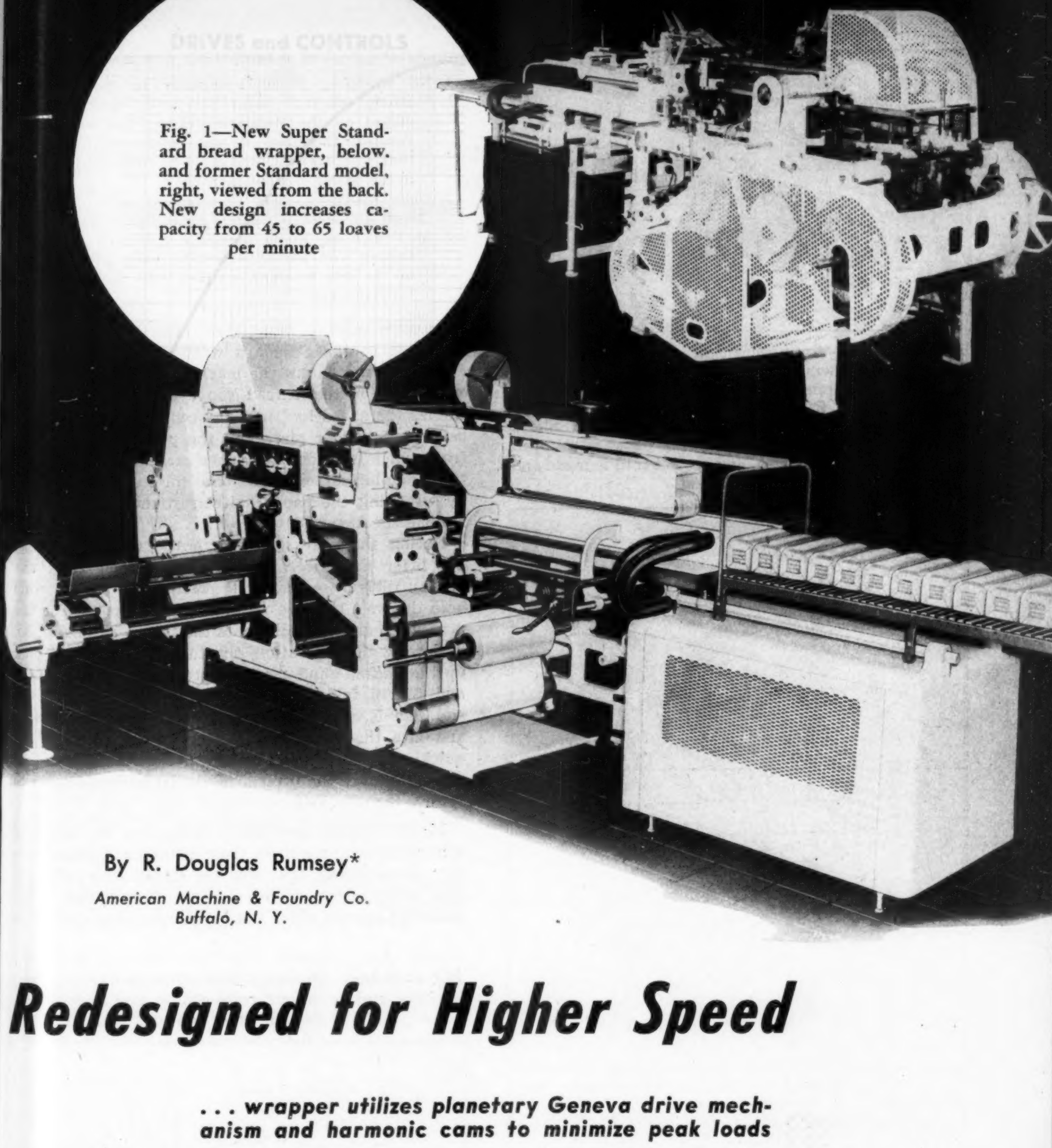


Fig. 1—New Super Standard bread wrapper, below. and former Standard model, right, viewed from the back. New design increases capacity from 45 to 65 loaves per minute

By R. Douglas Rumsey*

*American Machine & Foundry Co.
Buffalo, N. Y.*

Redesigned for Higher Speed

. . . wrapper utilizes planetary Geneva drive mechanism and harmonic cams to minimize peak loads

PROBABLY no other machine in a modern bakery equals the bread wrapper in complexity of its drives and controls. Because of variations in the infeed material and because of its special nature, the design of a wrapper involves many interesting and challenging problems. Although design of the American Machine & Foundry's Standard wrapper was still basically sound, a faster machine was needed. It was apparent that either a new machine or a redesign of the old machine was required.

In view of the excellent service record of the standard wrapper, it was decided to redesign it, increasing its speed from 45 to approximately 65 loaves per minute, *Fig.* 1. This approach had several advantages

* Now associated with Stanley Aviation Corp., Buffalo, N. Y.

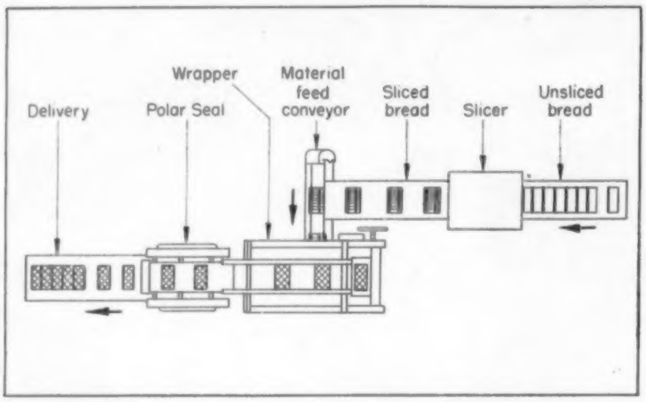
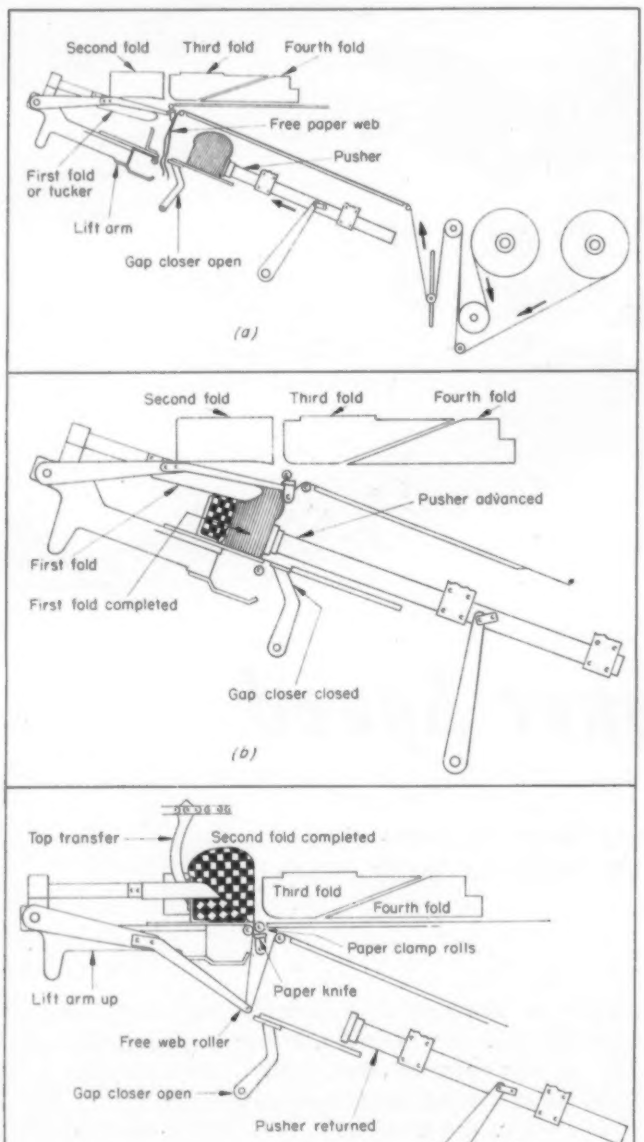


Fig. 2—Above—Arrangement of component units in Standard bread wrapper including slicer, material feed conveyor, wrapper, sealer, and delivery conveyor

Fig. 3—Below—Three steps in wrapping cycle. Sliced loaf is carried into the wrapper at *a*, paper is wrapped half way around loaf at *b*, and second fold is made and paper cut off at *c*



over development of a completely new machine:

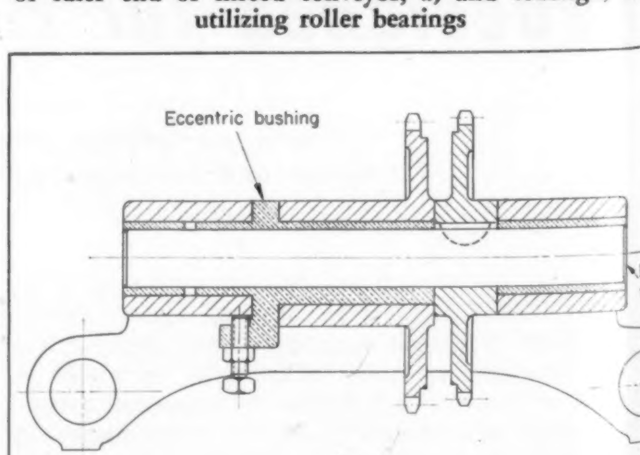
1. No wrapping principle superior to the "Armstrong demand" paper measuring system employed on the standard wrapper has been developed; therefore, it was considered unlikely that a superior method could be discovered within the course of a few months
2. Older machines could be modified, with conversion kit parts, if deemed economically advisable
3. Modernization would require a minimum of new tooling and parts, permitting early production of the new machine, coupled with desirable interchangeability.

Basic Wrapping Principle: The basic principles employed in the Standard wrapper can be seen in *Fig.* 2. Bread enters the machine from a slicer, and enters the material feed conveyor. This conveyor carries the sliced loaf into the center of the machine between a pusher plate and the paper web, *Fig. 3a,* from which position the pusher carries the bread into the wrapping material and onto the lifter arm, thereby wrapping the paper halfway round, *Fig. 3b,* and making the first fold.

During the next sequence in the wrapping cycle, the lifter arm rises, forming the second fold; the pusher returns to its dwell positon, the clamp rolls hold the web, the free web roller draws paper for the next wrap and the knife cuts the paper, as in *Fig. 3c.* Following this operation, the top transfer arms pick up the package and carry it through the heated third and fourth folders, past the end labeler where the end seal is applied, through the heated side sealers and into the Polar Seal delivery where the wrap is quickly cooled to set the adhesive. The instant the package leaves the lifter arm, the latter drops down to receive the next loaf.

Modifications Required: Increasing the speed of a machine requires more than installing a larger motor; nevertheless, that was the initial step taken on this development. Short tests were conducted at accelerated speeds which disclosed many weaknesses in

Fig. 4—Below—Drawings showing former arrangement of idler end of infeed conveyor, *a*, and redesign, *b*, utilizing roller bearings



the machine not evident from a study of service replacement part frequency records. These investigations revealed that development work would be required along the following lines:

1. *Machine movements:* Major cams and the Geneva drive chattered excessively at higher speeds. The original cam contours were generated by tangent circular arcs resulting in high shock loadings at the tangent points
2. *Wearing, weak or noisy parts:* Certain mechanisms demonstrated structural or other unworthiness when operating at greatly increased speeds
3. *Paper handling:* The original paper handling devices resulted in crushed bread, torn webs and excessive rewraps when overspeeded
4. *Appearance and control:* Modernization necessitated restyling, coupled with simplified control and adjustment
5. *Cost:* Design simplification was necessary to offset higher priced parts.

MATERIAL FEED CONVEYOR: The material feed to the wrapping machine is accomplished by a pair of roller chain conveyors running parallel to each other. On each chain, a series of pusher flights are mounted which support and convey the sliced loaf into the machine. The two chains can be adjusted differentially to accommodate changes in loaf length. This conveyor was found to be a source of rapid wear, excess vibration and malfunction when operated at high speed. At speeds over fifty loaves per minute, bread slices were disarranged and above sixty per minute, unsliced loaves jumped into the air. This trouble was in part attributable to the type of Geneva intermittent drive employed, as well as to the conveyor itself which overloaded the Geneva.

Higher speed with smoother action was a "must" for this conveyor and, in order to achieve it, lower

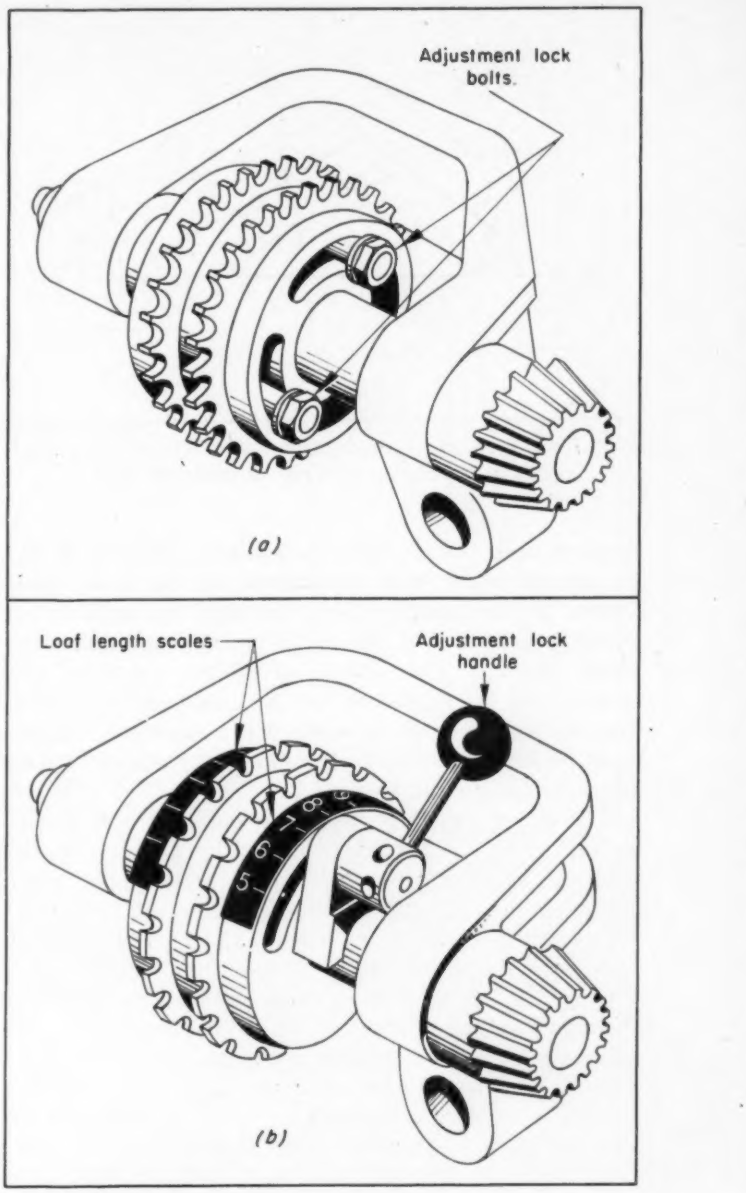


Fig. 5—Above, right—Drive end of old conveyor shown at *a*. Redesign, *b*, replaces bushings with antifriction rollers, includes adjustment locking handle

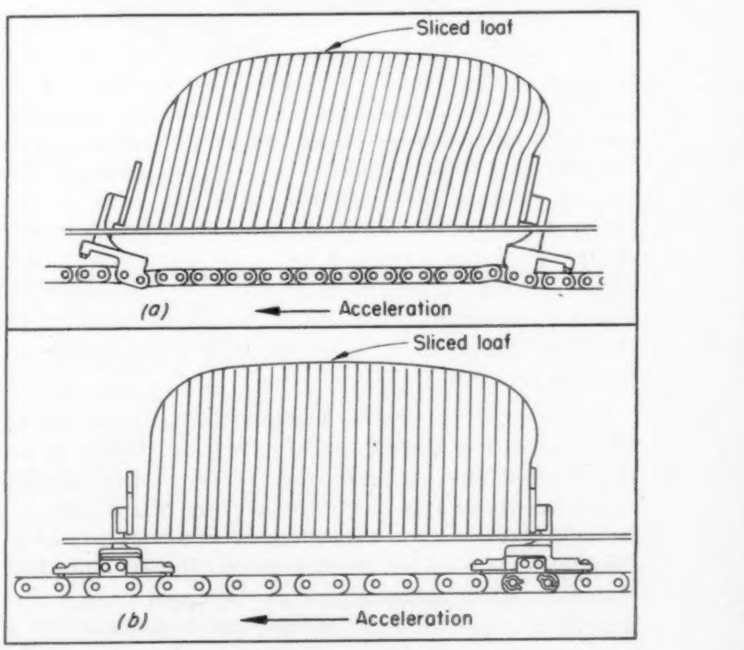
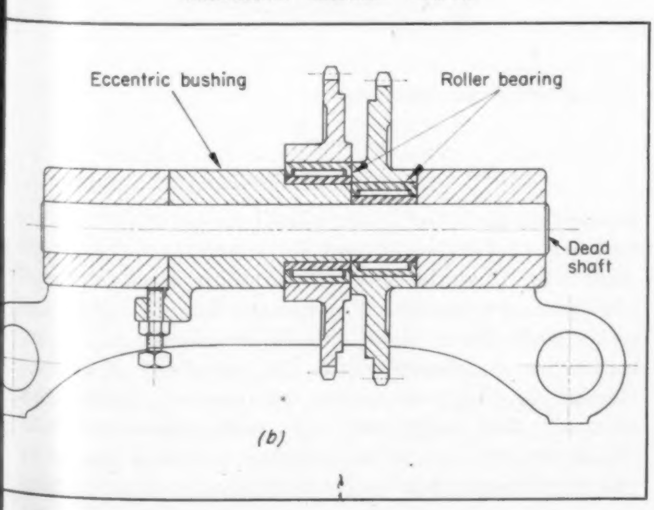


Fig. 6—Right—Old and new conveyor chain pusher flights shown at *a* and *b*, respectively. Redesign with longer links and guides has provided less weight and decreased friction forces



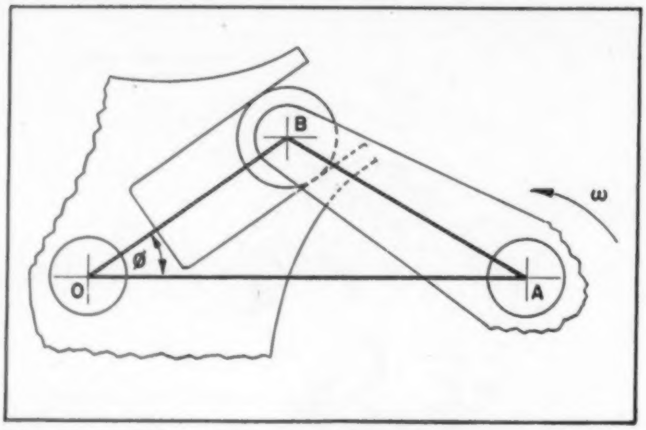
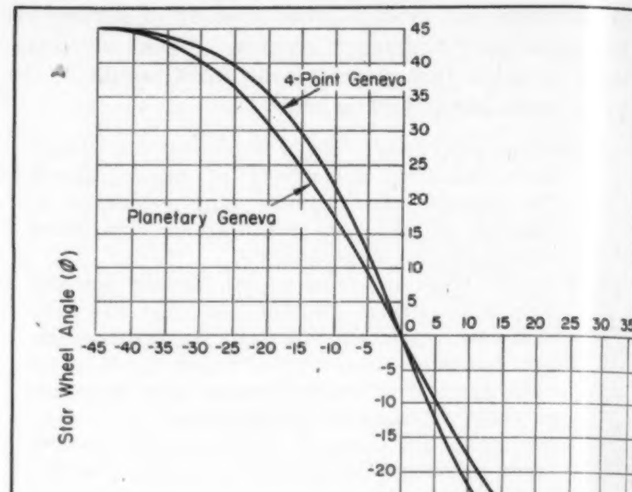


Fig. 7—Sketch of Geneva with nomenclature for analysis of mechanism of any number of slots



friction and inertia were necessary. The idler end of the conveyor was redesigned by replacing four bronze bearing surfaces, *Fig. 4a,* with two antifriction roller bearings, *Fig. 4b,* while retaining differential chain takeup. The drive end of the conveyor was modernized by replacing its two bushings with antifriction rollers and substituting a knobbed adjustment locking handle for the two bolts originally employed, *Figs. 5a* and *5b.*

The conveyor chain was analyzed next. A weight saving of 1.6 pounds was realized by going from the original standard roller chains to extended pitch roller chains, with no sacrifice in strength. The original pusher flights were bronze and served as chain links, which wore rapidly, *Fig. 6a.* These were changed to aluminum castings riveted to standard attachment links, *Fig. 6b,* thereby preserving the chains structural integrity and at the same time saving an additional 23 pounds or 62 per cent, making a total reduction of inertia loads of 33 per cent, including the bread, sprockets and shafts.

Chain Bending Increased Friction

The bread load is applied approximately two and one-half inches above the chain centerline, which caused considerable canting of the original pushers, resulting in increased chain tension. During acceleration, this chain tightening effect increased the system friction by more than 100 per cent in the original design. The new pushers eliminate this difficulty by being attached to chain links which are twice as long. The design also incorporates stabilizing extensions on each end. The overall gain with the new design is a reduction in driving torque of over 60 per cent.

MATERIAL FEED DRIVE GENEVA: The geometry of the standard wrapping machine is such that it is desirable that the material feed conveyor drive period equal the dwell time. The reason for this is that although the conveyor moves eighteen inches per feeding cycle (two cycles being required to carry the loaf from the slicer transfer into the wrapper) the transfer from the conveyor to the lifter is done by a pusher moving nine inches forward and back again. The original drive was by means of a standard Geneva lock mechanism consisting of a four-slot starwheel and double roller arm which provided equal drive and dwell times.

Many articles and papers have been written on Geneva mechanisms, therefore the data presented hereinafter will be kept brief and relevant. A general Geneva (any number of slots) can be analyzed as follows, where A is the arm center, O is the starwheel center, and B the arm roller, *Fig.* 7. If ω = arm rotation velocity, θ = arm position, ϕ = starwheel position, and $\alpha = OA/AB = \sqrt{2}$ on the 4-point starwheel, then at any instant:

Starwheel displacement is

$$\phi = \tan^{-1}\frac{\sin\theta}{a - \cos\theta}$$

Starwheel velocity is

$$\frac{d\phi}{dt} = \left(\frac{a\cos\theta - 1}{a^2 - 2a\cos\theta + 1}\right)\omega$$

Starwheel acceleration is

$$\frac{d^2\phi}{dt^2} = \left[\frac{a\sin\theta\,(1 - a^2)}{(a^2 - 2a\cos\theta + 1)^2}\right]\omega^2$$

These functions for a standard four-point Geneva are graphed in *Fig. 8a, 8b* and *8c* in comparison with the results that would be obtained with constant acceleration and a four-point planetary Geneva. It can be seen from *Fig.* 8 that both peak velocity and acceleration encountered with the standard four-point Geneva are high compared to constant acceleration motion. The rapid wear generally associated with these Genevas can be seen in *Fig.* 9 and is caused by the high torque forces (proportional to acceleration)

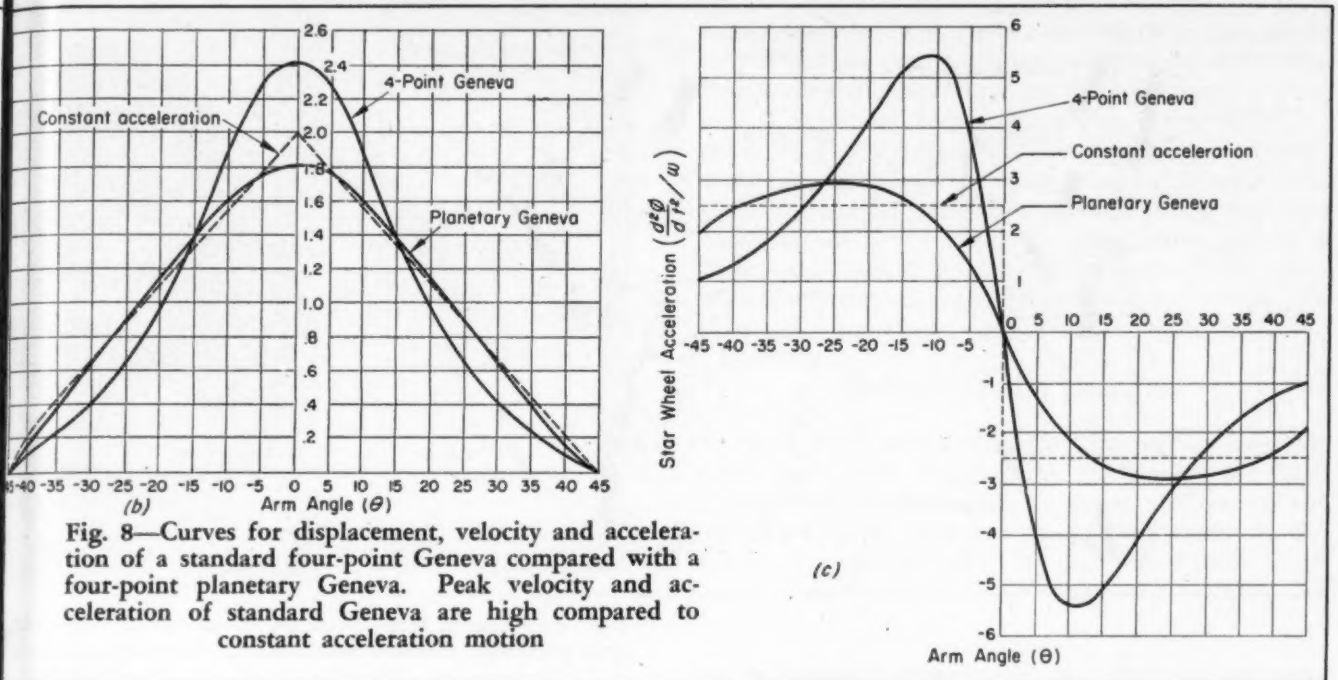


Fig. 8—Curves for displacement, velocity and acceleration of a standard four-point Geneva compared with a four-point planetary Geneva. Peak velocity and acceleration of standard Geneva are high compared to constant acceleration motion

combined with the short effective radius of the arm roller at the position of maximum acceleration, *Fig.* 10*a*.

As a means of overcoming the weaknesses of the standard Geneva and to provide an indexing mechanism that can be substituted on existing wrapping machines in the field, the unique planetary Geneva, *Fig.* 10*b*, was designed. The construction of this device is shown in *Fig.* 11. It may be noted that its effective radius, *Fig.* 10, is 0.75*R*, compared to 0.44*R* for the standard Geneva which, coupled with its lower peak acceleration, makes the maximum roller loads only one-third as high.

Thus,

$$\frac{\dfrac{0.44R}{5.42\omega^2}}{\dfrac{0.75R}{2.9\omega^2}} = 31 \text{ per cent track load}$$

where $5.42\omega^2$ is the peak acceleration of the planetary Geneva, as shown in *Fig.* 8.

An additonal advantage of the planetary Geneva over most other indexing mechanisms lies in the fact that the shaft center distances can be decreased by means of an eccentric bushing to compensate for wear in the lock without impairing the entry and exit characteristics of the roller.

CAM DESIGN: A great many articles have also been written on the subject of cam designs, therefore only the major factors will be mentioned. The original cams employed on the wrapper were essentially gravity or constant acceleration cams, closely approximated by circular arcs. These cams performed satisfactorily at speeds up to forty-five cycles per minute, but at speeds of eighty and higher, they destroyed themselves in a few days, *Fig.* 12. This was readily understandable because cam loadings vary with the square of the speed. The basic requirements for high speed cams are:

1. No abrupt changes in load
2. Minimum driven mass
3. Maximum period of operation
4. Maximum accuracy of contour
5. Maximum pressure angle and load noncoincident
6. Minimum system backlash
7. Low peak accelerations.

Of the commonly used cam contours,* gravity, constant acceleration or parabolic, cubic and harmonic

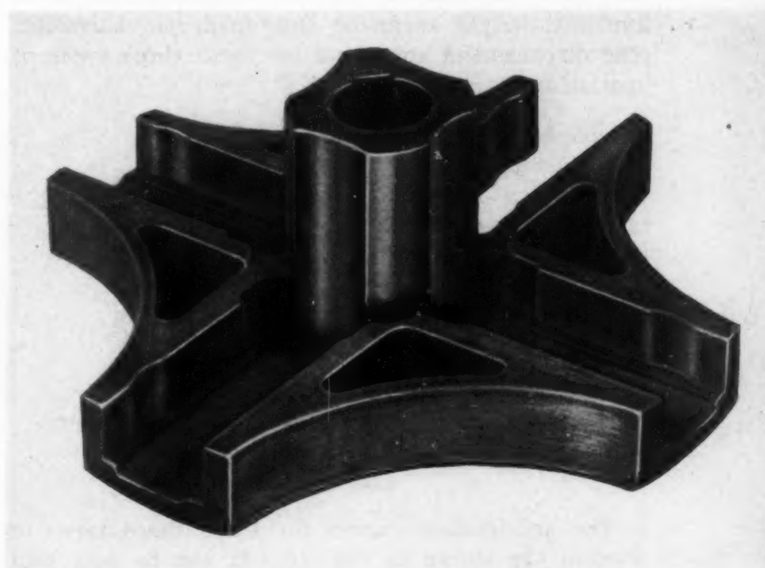


Fig. 9—Starwheel for standard four-point Geneva. Excessive wear was caused by high torque and short effective radius of arm roller

* References 1, 2 and 3 at the end of the article discuss the characteristics of various cam contours.

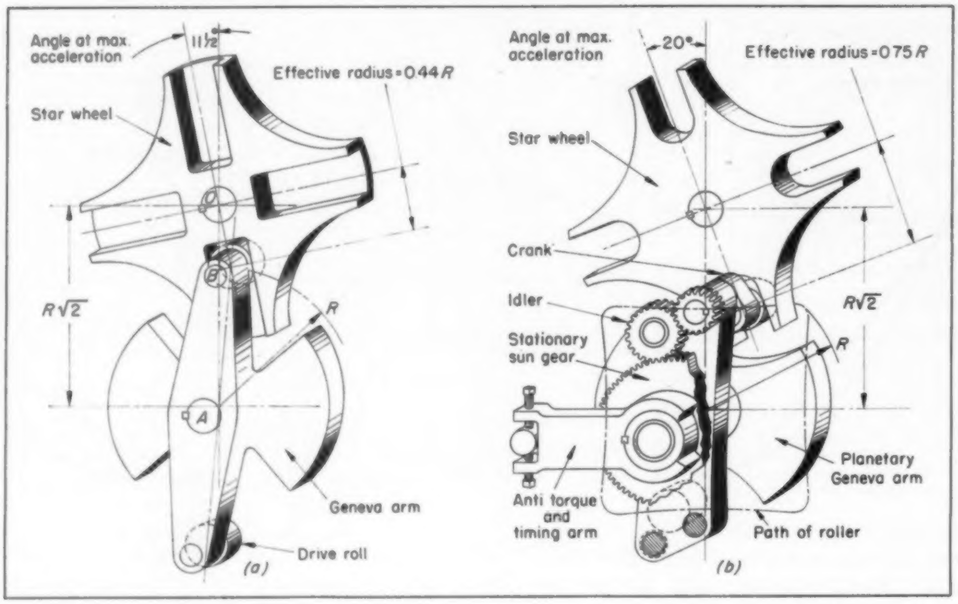
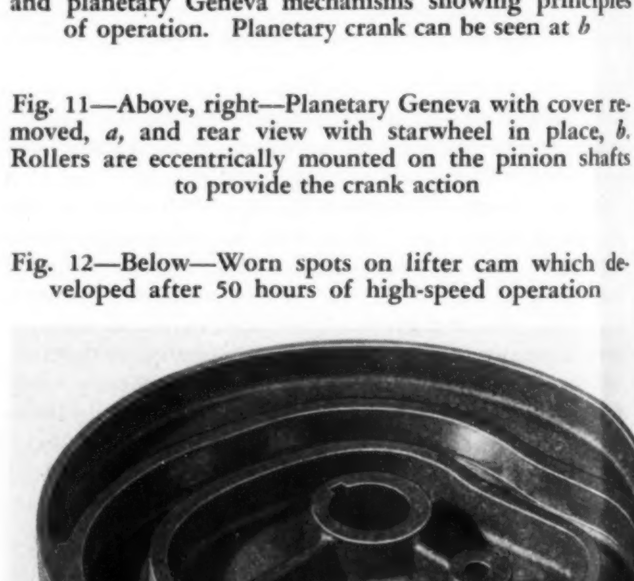


Fig. 10—Above, left—Sketches of standard four-point and planetary Geneva mechanisms showing principles of operation. Planetary crank can be seen at *b*

Fig. 11—Above, right—Planetary Geneva with cover removed, *a*, and rear view with starwheel in place, *b*. Rollers are eccentrically mounted on the pinion shafts to provide the crank action

Fig. 12—Below—Worn spots on lifter cam which developed after 50 hours of high-speed operation



contours are definitely unacceptable because of the abrupt load changes present. Motions that provide a smooth application of load are double harmonic or cycloidal, triple harmonic and quadruple harmonic. The displacement equations for these three types of motion are:

Double harmonic (cycloidal):

$$S = \frac{K_1 t}{2\omega} - \frac{K_1}{4\omega^2} \sin 2\omega t + C_1 t + C_2$$

Triple harmonic:

$$S = \frac{2K_2}{3\omega^2} \sin \omega t + \frac{K_2}{9\omega^2} \sin^3 \omega t + C_3 t + C_4$$

Quadruple harmonic (double cycloidal):

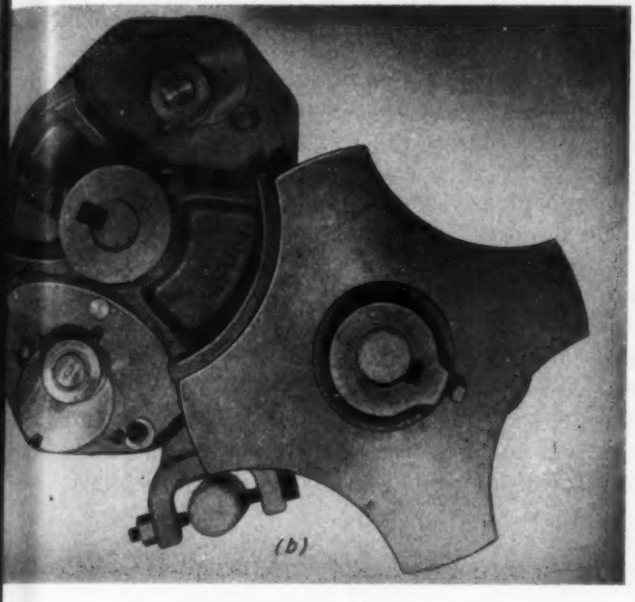
$$S = \frac{K_3 t^2}{2} - \frac{K_3}{4\omega^2} (1 - \cos^2 \omega t) + C_5 t + C_6$$

The acceleration curves for these three types of motion are shown in *Fig.* 13. It can be seen that with the triple and quadruple harmonic motions, the rate of change of acceleration is zero at the beginning, end and crossover point of the stroke. Hence, these motions are more capable of accommodating system backlash without chatter than double harmonic. Quadruple harmonic motion is superior to triple because its peak acceleration is low and its acceleration curve, not being as steep, is less likely to excite natural frequency vibrations in the system.

Combination of Motions Used

A combination of the quadruple and double harmonic motions are utilized on the new wrapping machine. Typical of these are lifter and pusher cams whose acceleration curves are shown in *Fig.* 14. It can be seen that the pusher uses a slow forward stroke, to prevent bread crushing, and a quick return. The lifter has a high upward acceleration with a low deceleration which reduces the tendency for the loaf to be thrown into the air. Various other combinations of these motions can be employed to suit particular applications.

The motion of the driven mass and cam contour will only coincide when the mass is directly connected to a radially moving cam follower. Since this condition seldom exists on actual mechanisms, computing the actual cam contour, back through the connecting linkages from the desired motion of the driven mass, frequently becomes a laborious mathe-



matical problem. As an alternative, the cam contour can be generated by physically duplicating the actual linkages on a master cam generator. A device of this type, which proved successful on the wrapper redesign, is shown in *Fig.* 15. On this generator, the master cam is scribed every two degrees with a line 0.002-inch wide, forming a circle of the follower's diameter. The desired position of the driven end of the system may be set with a micrometer from a fixed reference point. The completed master then is filed tangent to the lines, giving an overall accuracy of plus or minus 0.001-inch. After the master cams are completed, they are carefully measured and the results plotted on a large scale to check for possible errors. These same polar co-ordinate dimensions are tabulated on the cam drawings for shop inspection and use.

Unless extreme accuracy is maintained in generating cams of this type, the value of the contour form is lost because shock loads resulting from tool marks or inaccuracies can exceed the peak acceleration loads. Maintaining the necessary degree of contour accuracy results in costly cams. Therefore, as a means of minimizing this problem, experiments are currently under way utilizing resiliently mounted follower rolls which show good promise of success. It has also been found beneficial to employ barrel-shaped followers in track cams, where torsional deflections of the roller mounting lever cause cylindrical rollers to scrub both sides of the track.

Electrical System Simplified

Other Improvements: Other noteworthy improvements have been made in the machine in the form of utilizing constant-strength beam sections in the lap roll system, reducing the inertia loads on the cam by over 50 per cent. The paper feed drive, originally stepped pulleys, is now infinitely variable. All

Fig. 13—Below—Follower acceleration curves for double, triple and quadruple harmonic motions. Combination of double and quadruple harmonic motions is employed in final design

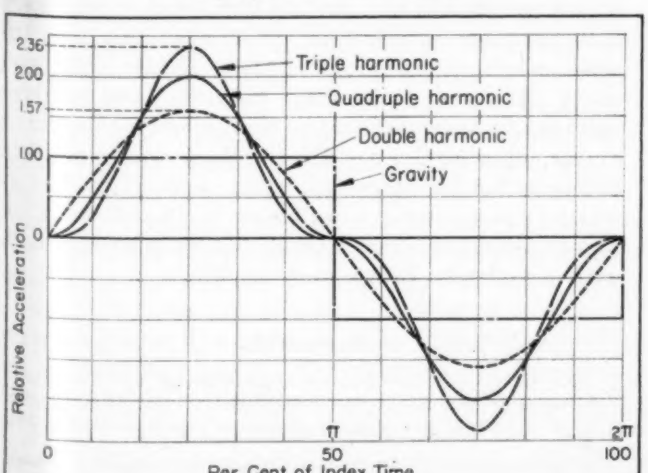
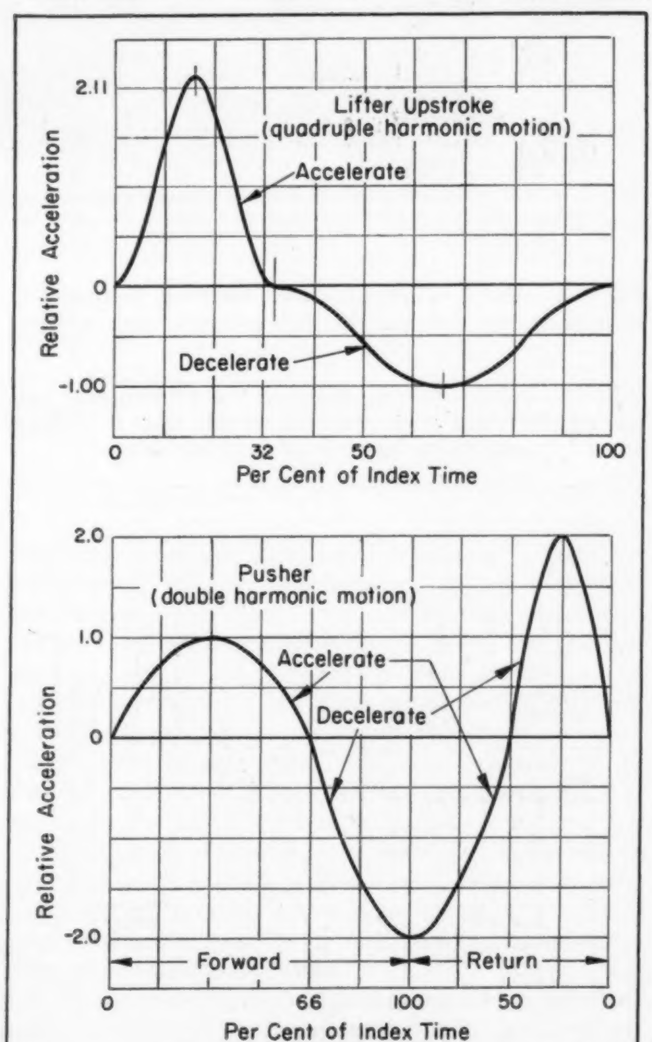


Fig. 14—Below—Lifter and pusher cam acceleration curves. Lifter has high acceleration, low deceleration. Pusher has slow forward stroke to prevent bread crush-
Pusher has slow forward stroke to prevent crushing



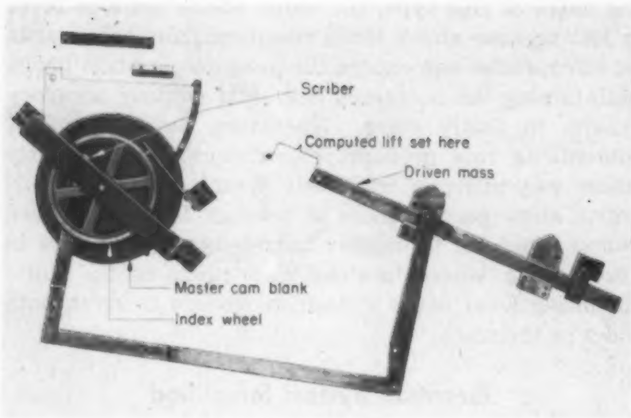


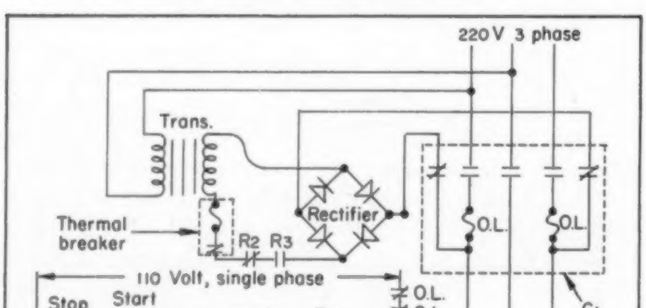
Fig. 15—Above—Master cam generator for producing wrapper cams satisfactory for higher speeds

the machine's guards and hand wheels have been restyled, as can be seen in *Figs.* 1 and 2.

The electrical system which originally covered the machine like a net was condensed into two compact units; the control panel, which now includes a tachemeter, *Fig.* 16, and the main junction box. Standard electric range switches and thermostats are used in this central panel. The machine formerly employed a manually operated friction clutch which was disengaged to permit quick stopping. Mechanical linkages and controls for this clutch were cumbersome and limited the number of operating positions. This clutch was eliminated by utilizing a plugging motor stop, permitting greatly simplified and condensed controls and greater flexibility as to location of the central equipment. Two forms of plugging were tested; full-reverse a-c and d-c plugging. Some difficulty in accurately timing the reverse plugging so the machine would stop fully without backing up gave preference to the d-c type, which is employed on the final machine. The wiring diagram for this circuit, which fully stops the machine in one second, is shown in *Fig.* 17.

Fig. 16—Above—Control panel showing tachometer, start and stop buttons, thermal controls, and switches

Fig. 17—Below—Wiring diagram for d-c plugging motor circuit which stops wrapper in one second without undesirable back up of machine drive



After redesign, the machine was field tested at 65 loaves per minute in a typical high-production bakery with complete satisfaction. Following this test, the same machine was subjected to a 25 per cent overspeed (80 loaves per minute) endurance test for three months, 24 hours each day. No major breakdowns or excess wear occurred. The machine was completely disassembled for inspection at the end of this test at which time a complete evaluation of the effects of the higher speeds on every part of the machine was made with completely satisfactory results.

References

1. John A. Hrones—"Key Factors in Cam Design and Application," Machine Design, April, May, June, 1949.
2. D. B. Mitchel—"Cam Follower Dynamics," Machine Design, June, 1950.
3. A. F. Gagne, Jr.—"Design of High Speed Cams," Machine Design, July, 1950.

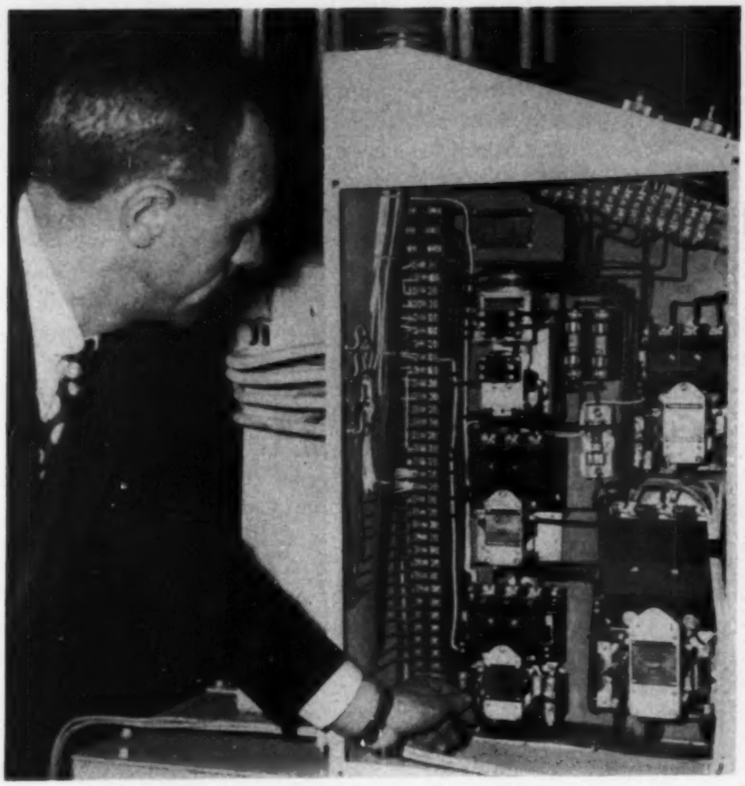


Fig. 1—Control for cement-block machine. Pneumatic timer, visible at top center of panel, is adjustable to control the process time

Timing Relays

. . . their selection and application in electrical control circuits

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SELECTING the best timing relay for a particular application, from the wide variety that are commercially available, entails recognition of the requirements of the job and a fundamental knowledge of the basic characteristics of each type. Since there are ten basic types of timing relays—escapement, motor, thermal, inductive, capacitor, electronic, dashpot, pneumatic, mercury, and inertia—the task becomes one of picking the relay with characteristics that match the particular needs of the application in question.

Because industrial processes are becoming more and more automatic, manufacturers are continually seeking methods to make these operations more dependable, consistent and foolproof. Often the application of a timer as an automatic control is all that is required. As machines become more complex, time-delays may be utilized to advantage in simplifying an otherwise complicated control by assuring proper sequencing of related mechanisms. In our everyday life automatic devices such as washing machines, dishwashers, ranges, molding presses and die casting machines are typical applications for such relays.

A timing relay can be considered as any device that produces a controlled time-delay period in an electrical circuit after a signal has been imposed on it. Manual signals could involve the manipulation of a dial, lever, or latch. Electrical signals could be the energization or de-energization of an electromagnet, coil, motor, or circuit of the timing relay. At the elapse of the predetermined time, electrical contacts open or close to connect or disconnect automatically some other electrical device.

If an electrical control is to feature a time interval and automatically initiate some other function, it must use a timing relay. In many cases, the selec-

tion of a timing relay is more or less evolutionary. First one type is tried, then another, until finally a "best relay" is discovered. The most effective approach, however, is to understand thoroughly what the needs are; that is, timing range, timing adjustment (local or remote); accuracy (repeat time and reset time); reliability of operation; size and construction; and sensitivity to variables such as voltage, temperature, atmospheric conditions, corrosion, dust, vibration, and shock.

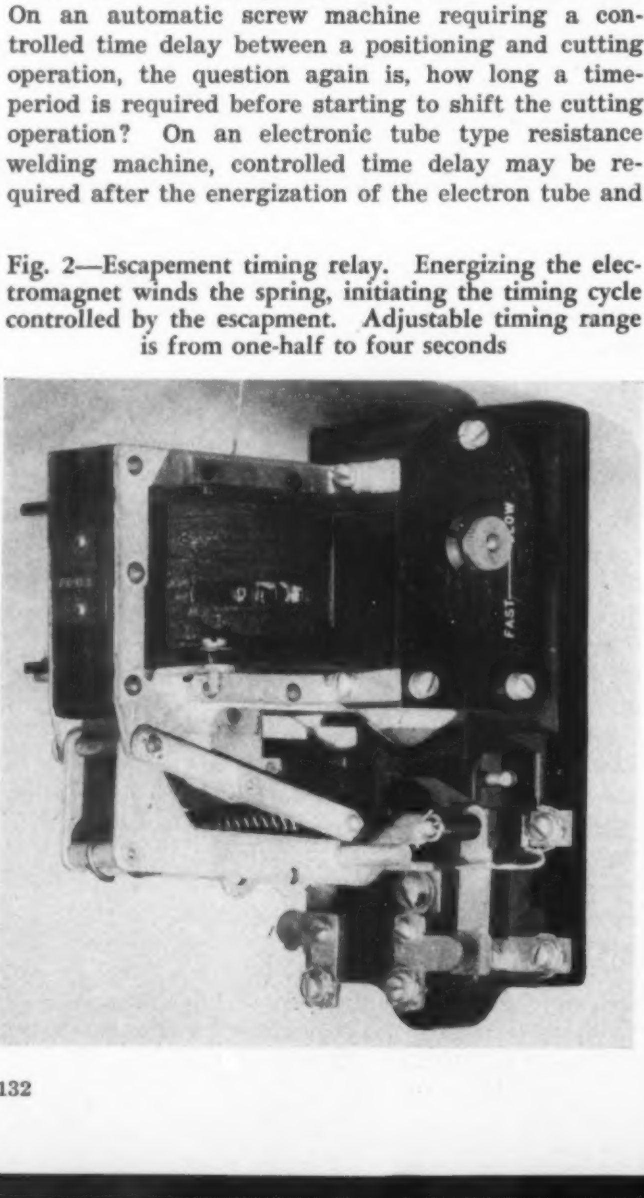
Selecting Timing Range

Before the designer can specify a timing relay to control the heat time on a plastic-molding machine, for example, he must know how long the heat is to be applied to the mold during the heating or curing process. If the heat time is to be thirty minutes, then a relay with that range must be used. On the other hand, if various mold sizes are to be handled on the machine—with the smallest mold requiring a ten-minute heat-time and the largest mold requiring fifty minutes—the timing relay must be capable of producing timing periods adjustable within these extremes. The same reasoning applies on a cement-block making machine using a timing relay, *Fig.* 1, to control the setting-time of a block in the mold. On an automatic screw machine requiring a controlled time delay between a positioning and cutting operation, the question again is, how long a time-period is required before starting to shift the cutting operation? On an electronic tube type resistance welding machine, controlled time delay may be required after the energization of the electron tube and before the application of power to the tube plate circuits. In this application a fixed time period is satisfactory since no adjustments or timing changes are necessary.

If the time period must be adjustable, the method of adjustment should be determined by the operating conditions. On the molding machine for example it may be desirable to have someone other than the operator who loads and unloads the molding machine, make the timing adjustments. The set-up man who would be responsible for inserting the correct mold could be responsible for setting the heat-time. In this case, the timing adjustment should be on the relay itself, and the relay located in a cabinet to insure against unauthorized changing of the time period.

For a mixing machine, however, the operator usually controls the time-delay depending on the ingredients being mixed or blended. One batch or solution may require 30 seconds and the next batch 60

Fig. 2—Escapement timing relay. Energizing the electromagnet winds the spring, initiating the timing cycle controlled by the escapment. Adjustable timing range is from one-half to four seconds



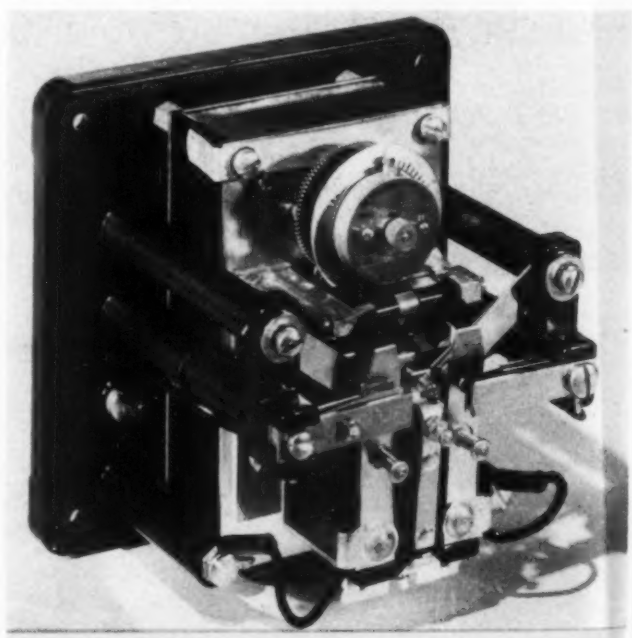
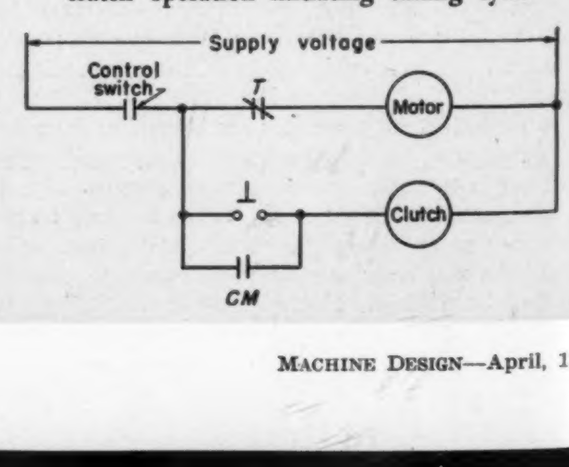


Fig. 3—Above—Synchronous-motor operated timing relay having three adjustable timing ranges up to 100 minutes. A magnetically operated clutch permits initial running of the motor to reduce inertia errors in timing

Fig. 4—Below—Schematic diagram of motor timer showing how motor is energized prior to clutch operation initiating timing cycle



seconds. If the timing relay and other associated controls are built into the base of the mixer, it would be inconvenient to adjust the relay unless a mechanical connection is used between the timing relay in the base and an indicating dial conveniently located on the outside of the machine. The typical X-ray machine has a timer dial on the outside of the cabinet, linked to the timer mechanism located inside the housing or cabinet to provide convenient adjustment.

If the relay must be mounted in a location inaccessible to the operator then a remote type of adjustment should be utilized. On some automatic arc welding machines, for example, the controls are remote from the operator's stand, yet the operator must make timing adjustments. Also, in hazardous locations, it may be economical to use controls in standard enclosures, rather than in explosionproof housings, and mount them in an area remote from the hazardous area. If adjustable timing relays are associated with the control, a relatively inexpensive explosionproof cabinet, containing a rheostat, can be mounted in the hazardous area and connected to the timer and other controls located in a nonhazardous area.

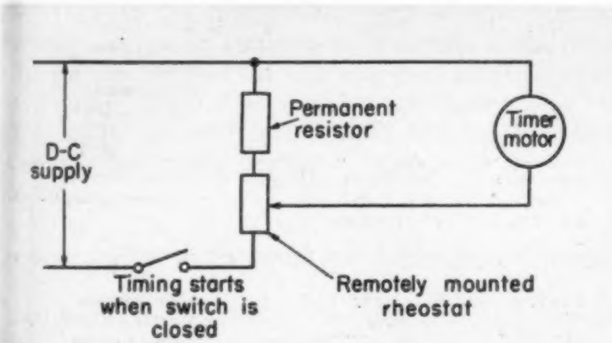


Fig. 5—Above—Direct-current motor timer connected for remote operation. Adjustment of the rheostat controls the voltage to the motor, producing a corresponding change in timing. A permanent resistor is connected in the circuit to insure minimum torque requirements of the motor

Fig. 6—Below—Bimetal timer for a fluorescent tube starter. When voltage is applied, an arc forms between the U-shaped bimetal strip and the single rod, heating the strip until it bends sufficiently to touch the rod and extinguish the arc. When the bimetal cools, the circuit is interrupted, completing the time delay. The condenser suppresses radio interference

The type of adjustment, with respect to the ease with which the timing period on the relay may be varied, is dependent on the operating circumstances. On some applications the turning of an adjusting screw on the relay is sufficient. On a planer, for example, a timing relay is sometimes used to introduce a dwell or time delay at the end of a cutting stroke to permit the cutting tool to automatically move or feed for the next cutting operation. It may be possible for the operator or set-up man to experiment with the timing adjustment to get a satisfactory delay corresponding to the time required to make the cutting-feed operation. On other applications such as a photo-printing timer, the operator wants a set time to be accurate and consequently the timing relay must have a conveniently-marked dial.

Determining Limits of Accuracy

Associated with ease of adjustment is the problem of timing accuracy. When an operator turns the dial to 10 seconds, how close to 10 seconds must the resulting period be? On the photo-printing timer, a variation of ± 10 per cent usually is tolerable, in that the resulting density of the photographic print made from an 11-second exposure would be as acceptable as a print made from a 9-second exposure. On a molding machine, the set-up man might adjust the

Fig. 7—Thermal timing relay. Bimetal strip is connected in the secondary of the current transformer. Time delay is controlled by gap adjustment

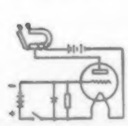
timing relay for a 10-minute heat time and actually get either 8 or 12 minutes. If he is on hand to observe the processing of several pieces this discrepancy would not be too serious.

Probably more important than adjustment accuracy, is the repeat accuracy. The timing period on a honing machine may vary ±10 per cent from piece to piece and still produce satisfactory results whereas a ±10 per cent variation in the heat-time of a molding machine would produce either burned or uncured pieces. The variation in timing that can be tolerated for a particular process then determines the accuracy that must be inherent in the timing relay.

After a timing relay has gone through its timing cycle or period, an interval called the reset time is required for the relay to assume its original condition prior to another cycle. This reset time is usually different for each type of relay. Referring again to the molding machine, the reset time can be as long as it takes the operator to remove a finished piece and reload the machine.

On highly automatic machines, however, the time required to reset a relay may be a factor in determining the maximum production output of the machine. For this reason the designer must decide if the resetting time of the relay will ultimately inter-

Table 1 — Characteristics and Features of Timing Relays

	Type	Operation	Ranges	Max Error†	Adjustment	Cost*	Application
	Escapement	Escapement retards operation of contacts	5 sec to 24 hrs	½ to 1 sec	Local	1	Time clocks X-ray timers Ovens Photographic printers
	Motor a-c d-c	Motor operates contacts through gear train	1 sec to 24 hrs	0.1-sec (a-c) to 5% (d-c)	a-c, Local d-c, Local or remote	1-2	General-purpose Battery chargers
	Thermal a-c d-c	Heater causes bimetal to bend, operating contacts	5 sec to 8 min	5 to 10%	Local or fixed	1	Light blinkers Fluorescent tube starters
	Inductive d-c	Heavy copper coil retards flux decay, delaying contact operation	0.1-sec to 4 sec	5%	Local or remote	2-4	Motor acceleration
	Capacitor d-c	Stored energy in capacitor delays drop-out of relay	0.1-sec to 50 sec	5%	Local or remote	2	General-purpose Motor acceleration
	Electronic a-c d-c	Tube amplifies energy from timing capacitor, controlling relay	0.02-sec to 3 min	1%	Local or remote	3-5	Resistance welders Machine tools
	Dashpot a-c d-c	Contact movement is retarded by dashpot	1 sec to 60 sec	5 to 10%	Local	1	Overload protection Motor acceleration
	Pneumatic a-c d-c	Contact movement is retarded by diaphragm	0.2-sec to 4 min	5 to 10%	Local	2	General-purpose Machine tools
	Mercury a-c d-c	Mercury contacts delayed by gas escapement	0.2-sec to 20 min	5%	Fixed	2	Filament heat-up timers General-purpose
	Inertia a-c d-c	Heavy weight retards contact movement	0.08-sec to 0.1-sec	5%	Fixed	1	Sequencing circuits

* Relative cost for single-contact timers, 1 being lowest cost.

† Plus or minus values.

fere with successful operation of his machine. Suppose a screw machine is designed to perform three distinct operations, with timing periods between each to allow for tool movements or shifting of the work. The designer may feel that proper operation should consist in pressing the start button and permitting the machine to go through its entire cycle. However, there is nothing to prevent an operator from ignoring the recommended operation stopping the machine anywhere in a timing cycle. Loss of voltage may also shut down the machine during the cycle.

The question then arises, should the timing cycle continue where it left off if the operator immediately restarts the machine or if the power supply resumes after a shutdown? Or should the control circuit be so designed that the machine will automatically restart a complete new cycle? If the latter is the answer, then the timing relays must be reset. If the reset time on the first timing relay is dependent on the machine going through its complete cycle, then it is conceivable that the operator may stop the automatic cycle before it is complete and attempt to restart before the first timing relay has had a chance to reset. Inaccurate timing or false operation would result.

Dependability is Essential in Automatic Controls

Naturally it is desirable to use a reliable timing relay. Some relays, however, are more reliable than others. Stopping a machine in a long chain of automatic production sequences would stop the entire production line. This could be a special conveyor or a machine performing one operation. Reliability then is of utmost importance. Unattended automatic devices fall in this category.

Allowable size of a timing relay depends on the space conditions existing in the ultimate location or in the machine. In most machines, space is at a premium. More and more automatic controls are being crowded in existing machine frames. Some applications, however, are not critical from the space viewpoint. Other problems associated with size and construction are maintenance and replacement of parts. A small and delicate relay would suffer under the abuses of a maintenance crew accustomed to handling everything with heavy gloves and large wrenches.

Variables such as voltage and temperature changes, atmospheric conditions, corrosion, vibration, and shock must be considered if present. Any device or machine requiring an attendant for its operation is usually in a location having a fairly constant temperature. Timing relays used on furnace-process controls, however, are exposed to high temperatures and must be operable at those temperatures.

On outdoor applications such as oil-well pump controls, a relay must be used that operates satisfactorily over seasonal temperature ranges. In outlying installations, due to long lines, the supply voltage and frequency usually vary, consequently timing relays more or less insensitive to these variations should be used. On vat quenching machines, the timing relays may be exposed to corrosive atmospheres, or dust conditions as in coal blending plants and steel mills.

After all the requirements have been carefully tabulated the choice of relay in most cases is straightforward. As shown in the comparative chart, TABLE 1, the list of applicable types of relays is first narrowed to those capable of giving the desired timing range. Type of adjustment narrows the choice still further, and so on. The chart itself is not all-inclusive but it does indicate the general trend of characteristics as they apply to basic designs.

If the relay characteristics needed are not found in any particular relay, a combination of two types or possibly special designs may answer the problem providing cost permits. If the designer must diverge from standard commercial designs, a thorough knowl-

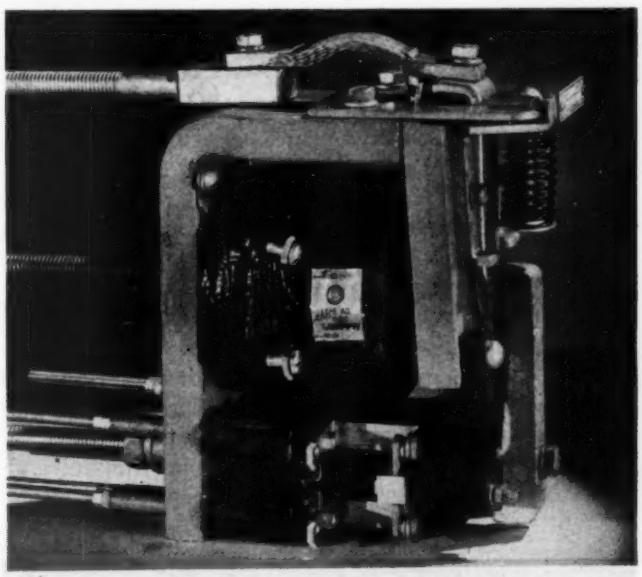
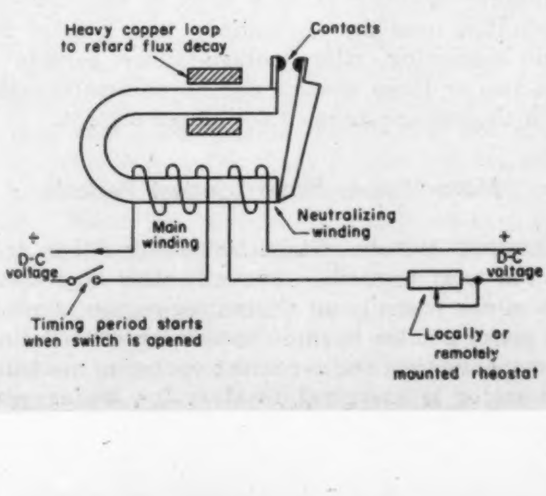


Fig. 8—Above—Inductive type timer for direct-current circuits. This relay has 150 ampere contacts and may be set for periods ranging from one-half to four seconds

Fig. 9—Below—Inductive timer connected for remote control of timing cycle. The rheostat controls the direct current flowing in a neutralizing winding, bucking the flux of the main winding, thus controlling the timing period



edge of the basic designs is a great help in the dedevelopment of a special timing relay. Features of each type of timing relay will be discused in the following:

Clock Type Timers Wind on Setting

ESCAPEMENT: This type of timing relay is a common and inexpensive device. The timing mechanism is similar to that employed in an alarm clock or similar time piece. This relay is utilized in timers for X-ray machines, photographic printers, process ovens etc. Usually the escapement mechanism is motivated by a spring which is manually wound. A dial on the timer may be manually rotated to a particular time delay desired—the turning of the dial automatically winding the spring. When the dial is released, the spring operates the necessary gear train in the direction to open or close the control contacts. The practical time delay range is approximately 5 seconds to 24 hours, the degree of accuracy usually being in terms of a fraction of a second.

Changes in temperature, causing expansion and contraction of the escapement mechanism, will affect the accuracy of mechanical timers. For a fixed temperature, a long time delay will have the same fraction of a second accuracy as a short time delay. In other words, the per cent error will be greater for shorter periods than for longer ones.

Escapement relays usually are inexpensive. Due to the manual operation, this class of relay does not receive severe or highly repetitive operation and hence holds up fairly well. Contacts usually associated with these devices have a current carrying capacity of approximately 10 amperes at 110-volts alternating current or ¼ ampere on 115-volt direct current.

Escapement mechanisms are used often on simple direct-current resistance type motor starters for timing the acceleration period. The escapement mechanism, however, is motivated by an electromagnet spring, *Fig.* 2. When the magnet coil is energized, a solenoid closes instantly to operate the holding circuit. At the same time it extends a heavy coil spring to actuate the time-delay contact mechanism, which is retarded in its motion by the escapement mechanism. The timing range on such a device is usually from ½ to 4 seconds.

Another example of this type is the time-delay pushbutton used on applications that require automatic restarting after voltage-failure periods less than two or three seconds and no automatic restarting if the voltage stays off for longer periods.

Motor Types Have Longest Periods

ELECTRIC MOTOR: These time-delay relays are by far the most versatile. Basically, this type consists of a motor (usually an alternating-current synchronous motor similar to those used in an electric-clock), a gear reduction, and a contact operating mechanism. The motor is energized to start the timing period. After a predetermined time interval has elapsed, the mechanism has moved or rotated sufficiently to operate the contact mechanism.

The electric motor relay usually has relatively good accuracy. For example, a 2/10 to 120-second timer may have an accuracy of 1/10-second. A 2-second to 20-minute timer may have a 1-second accuracy. A 1/5 to 120-minute timer may have an accuracy of 6 seconds. A 2-minute to 20-hour timer may have an accuracy of 1 minute. In other words, the timer is most accurate on its longest time setting. A timer with a 20-minute scale then can be used for an application requiring a 15-minute cycle as well as a timer with a 20-hour range. However, if accuracy is important, the timer with the lower range should be chosen for the application.

Clutch Reduces Timing Error

A loss in time, or error, usually occurs during the acceleration of the timer motor from standstill to synchronous speed. The percentage error is usually highest on time settings less than one or two seconds. This acceleration effect is reduced when using a timer with an automatic clutch mechanism to engage the motor, *Fig.* 3. The timer motor is energized before the timing cycle is initiated. Closing the clutch switch, *Fig.* 4, energizes its magnet causing the motor and contact mechanism to be engaged. After the time interval, contact *T* opens to de-energize the motor. The clutch magnet is kept energized through its contact *CM* until the voltage is interrupted by other contacts in the control scheme.

While the most common variety of electric-motor relay uses a synchronous type alternating-current motor drive, some types are driven with direct-current motors. The most notable application for a direct-current motor relay is one in which the timing period is to be remotely adjusted, *Fig.* 5. Since speed is proportional to the voltage, a rheostat can be conveniently used to adjust the voltage. Timing ranges can be obtained from 1 second to many hours. However, the accuracy of the relay will depend on the accuracy of the voltage supply.

THERMAL: The thermal or bimetallic timer is relatively inexpensive. Typical applications include signal blinkers, fluorescent tube starters, and overload relays. The fundamental principle of operation is the heating of a bimetallic strip in which the bending motion of the strip is used in operating a contact mechanism. Not all thermal relays, however, use bimetal strips. In a blinker light the expansion of a spring, due to current passing through the spring, sometimes operates a contact. The least expensive relays use an electric heater to heat the bimetal strip, *Fig.* 6. This is called the indirect-heated type. Practical timing ranges are from 10 seconds to approximately 5 minutes. This type of relay is rather inaccurate and is unusually susceptible to changes in ambient temperatures.

More accurate varieties are available in which the bimetal strip itself is in the heating-current circuit,

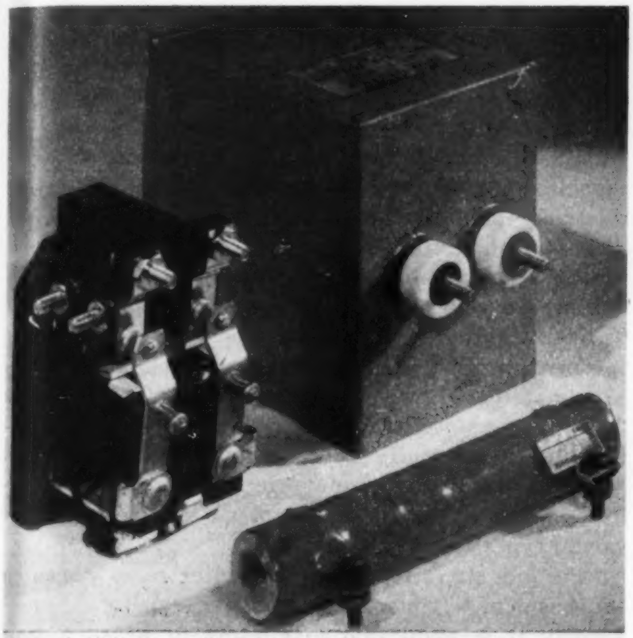


Fig. 10—Above—Capacitor timer components for direct-current circuits. Small percision relay may be mounted on the control panel and the more bulky capacitor may be mounted elsewhere. A resistor extends the timing period as desired

Fig. 11—Below—Connections for remote control of timing for capacitor type timer. By increasing the resistance in the discharge circuit the time period is lengthened

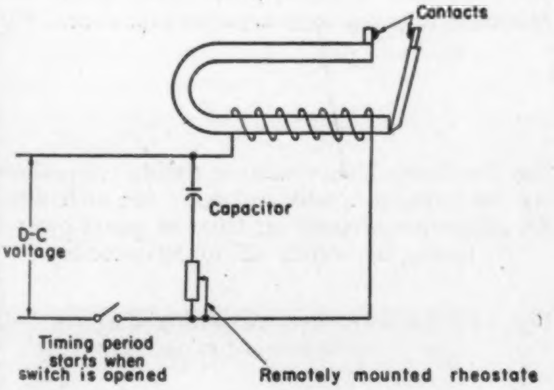


Fig. 12—Below—Simple modification of circuit for capacitor timer in *Fig.* 11. Second set of contacts on double-pole switch charges the capacitor direct, minimizing resetting time. Timing starts when switch is opened

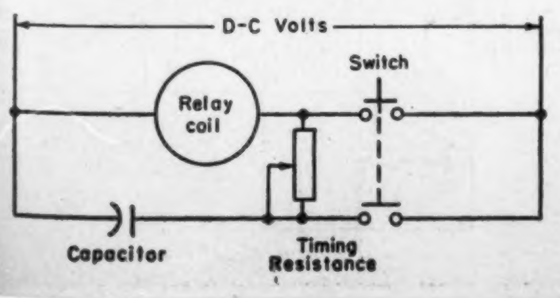


Fig. 7. In this way, heating is developed inside the bimetal itself rather than absorbed by convection. Called the direct-heated type, this relay usually has the bimetal strip in the secondary of the current transformer, limiting the relay to alternating-current applications. When voltage is applied to the primary of the transformer, a low voltage and high current is produced in the secondary circuit. This high current heats the bimetal strip to cause the bending. Having a timing range approximately 5 seconds to 8 minutes, this device does not work satisfactorily on highly repetitive operations because the reset or cooling-time varies from 10 to 50 seconds.

Some thermal timers are designed for immediate recycling. These employ the combined heating and cooling time of the thermal element before the final switching action occurs. In this way, the element is cool at the end of the cycle and ready to start a new timing cycle when required. Since the cooling time usually represents a larger part of the total time and since the cooling rate is independent of the voltage, the timing of these units is slightly modified by small changes in operating voltage. Practical timing ranges are approximately 15 seconds to 2 minutes. These relays have an accuracy of approximately 10 to 20 per cent.

May Utilize Both Heating and Cooling Time

Most bimetal devices operate the contacts directly, hence the contacts are of the slow-make, slow-break variety. Some relays, however, are available in which the bimetal is in the form of a cupped disk. Unequal expansion of the disk surface with increased temperature causes the bimetal to snap from one cup position to the other producing a snap-action contact.

INDUCTIVE: The flux-decay or inductive timing relay is used where short time delays of ½ to 4 seconds are required. This relay, *Fig.* 8, operates on direct-current and produces its time delay when the magnetizing coil is de-energized. Currents are induced in a highly inductive coil or in a heavy copper loop in the magnetic circuit, thereby retarding flux decay. In this way, the relay does not drop out when the voltage is interrupted but remains closed during a period determined by the effectiveness of the inductive-loop circuit.

The amount of energy producing the time delay is dependent on the size of the magnet on the relay. The applied voltage on the magnet coil determines the amount of flux, the maximum flux for a given magnet being the saturation of the magnet. It can be seen that variations in voltage will not produce great variations in flux if the magnet coil is designed to produce saturation with the lowest voltage anticipated. Since the amount or inductance required is high, the resulting timing ranges are of necessity fairly short, in the order of ½ to 4 seconds. By careful design and manufacture, these relays can be made accurate to approximately 10 per cent.

Making changes in flux can be accomplished with a neutralizing coil. Referring to *Fig.* 9, when the

main winding is de-energized, the heavy copper loop tends to maintain the flux in the relay. When the flux diminishes sufficiently, the relay armature drops out, opening the contacts. The neutralizing winding produces a flux in opposition to the main flux. The resultant flux is not as high and consequently, when the main winding is de-energized, the flux in the relay reaches the drop-out value sooner. The amount of neutralizing flux is controlled by the rheostat which may be mounted locally or remotely. This scheme is for timing ranges of ½-second to 4 seconds.

Metallic Rectifier Provides Direct Current

Most timing relays have control contacts with a capacity of approximately 10 to 15 amperes at 110-volt alternating-current. Inductive timing relays are also constructed with heavy contacts capable of carrying as much as 200 amperes. These are used to provide automatic acceleration timing and also to short out the resistor steps of a direct-current motor starter. The combination timer and contactor can also be used with wound-rotor motor accelerating controls. Rectifiers, usually copper oxide or selenium, are used to provide direct-current for the timer coil if direct-current is not available.

CAPACITOR: This type of timing relay employs a small direct-current relay and a capacitor. The two are connected in parallel so that when the voltage is removed, the charge built up in the capacitor is discharged through the relay. The time required for the relay coil to discharge the capacitor to the point where the relay drops out is the time delay. This scheme differs from the inductive timer in that the energy producing the time delay comes from the capacitor. To produce long delays on the inductive time relay, the relay itself becomes large, whereas on the capacitor type, the relay itself remains small and the capacitor becomes larger. By the use of a high-resistance coil or a high resistance in series with the capacitor, to retard the capacitor discharge rate, a relatively long time delay can be obtained, *Fig.* 10. This series resistance may be a rheostat to provide timing adjustment. *Fig.* 11 illustrates a typical circuit where the rheostat may be located remotely if desired.

Since it takes a relatively large capacitor (approximately 200 mfd) to obtain about 50-second time delay in conjunction with a sensitive relay, the combination is normally used on shorter time periods only. Although large low-priced electrolytic capacitors are available, they are not as reliable as paper-wound types.

Like inductive timers, the capacitor relay works on the basis of a time delay on de-energizing. Consequently it is applicable to circuits in which a time delay is desired if the voltage fails. The scheme can be used with a standard motor starter in the circuit to provide automatic restarting after a power interruption of a few seconds. For example, a very short power interruption will not cause a pump to slow down excessively. After several seconds delay the relay can be made to drop out, disconnecting the motor and thus assuring recycling of the starting device in the event that too long a time delay has elapsed during loss of voltage.

The reset time of a capacitor relay is determined by the time it takes to charge its capacitor. *Fig.* 11

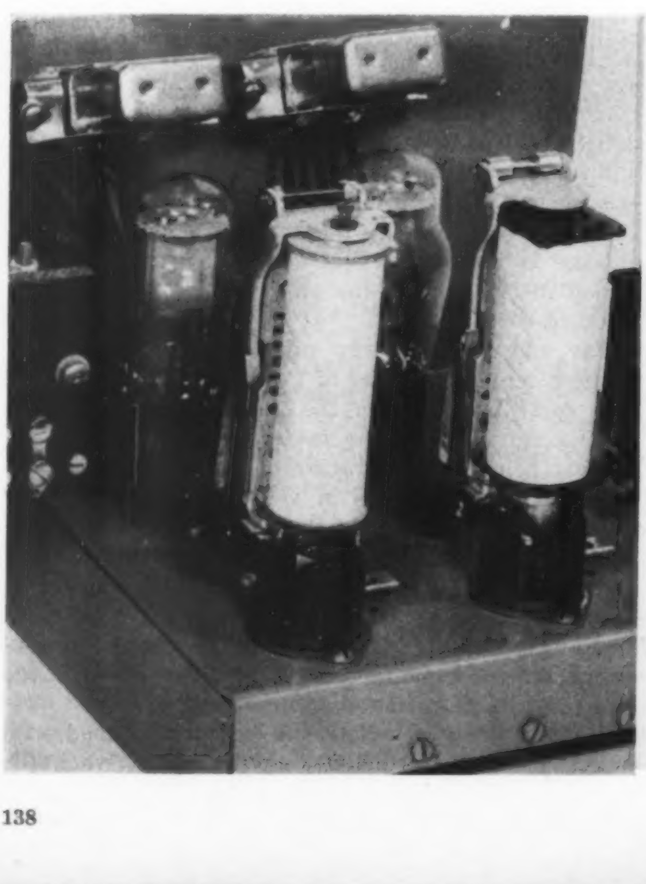
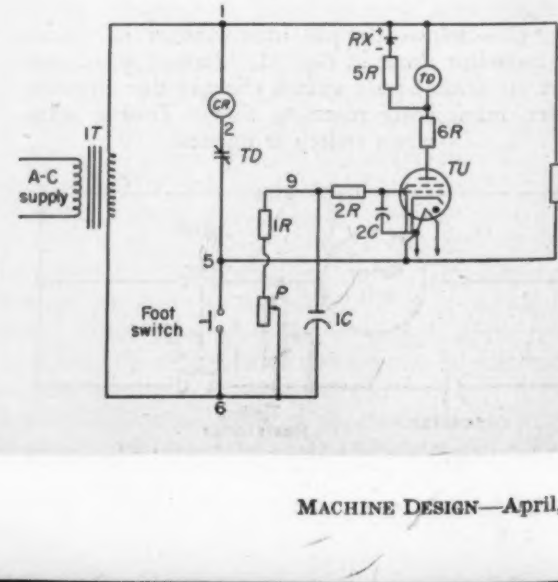


Fig. 13—Left—Two electronic timing relays showing the capacitor, tube and relay for each timer. An adjustable rheostat on front of panel controls timing accurately up to 30 seconds

Fig. 14—Below—Schematic wiring diagram for a typical electronic timing relay



illustrates a simple circuit using an adjustable resistance to vary the time period. When the initiating switch is opened, the capacitor discharges through the relay coil and resistance. As the capacitor and resistance both become larger to produce longer time delays, the charging time or reset time increases. *Fig.* 12 shows a simple reconnection of the circuit to minimize resetting time. Here, full direct-current voltage is applied across the capacitor to speed its charging. On the basis that the capacitor always discharges at the same rate after being charged to the same voltage, the accuracy of timing is dependent on the repeat drop-out accuracy of the relay itself.

ELECTRONIC: The electronic type time-delay relay, *Fig.* 13, is fundamentally a capacitor-discharge relay since the time delay of a capacitor discharging through a high resistance is used. The main difference lies in the fact that an electronic tube, acting as an amplifier, uses the discharging timing characteristics of the capacitor to operate a small relay. This means that the capacitor does not supply the energy to operate the relay direct and as a result can be much smaller. On most electronic timers, the timing capacitor is connected in the grid circuit of the electronic tube, and the control relay is connected in the plate circuit of the tube.

Capacitor Is Charged Automatically

In standby condition, the timing capacitor maintains a charge and is ready for the timing cycle or period. When the timing period is initiated, the capacitor starts to discharge through a high resistance and continues to discharge until the voltage across it reachs a predetermined value. When a gas thyratron tube is used in circuit, this voltage is the breakdown voltage of the tube. At this time, the tube starts to conduct to energize a relay, terminating the preselected time period. Operation of the contacts on the relay recharges the capacitor (this is called the reset period) and it is automatically ready for the next timing cycle.

In *Fig.* 14 is shown a diagram for a typical electronic timing relay. Capacitor 1*C* is normally charged with terminal 6 positive and terminal 9 negative through the following path: from the transformer lead through capacitor 1*C*, through resistor 2*R*, through the grid cathode of tube *TU*, through resistor 3*R* to the transformer terminal. Grid of tube *TU* acts as the anode of a half-wave rectifier for this charging action. Closure of the foot switch prevents further charging of this capacitor. It also energizes relay *CR*. The charge of capacitor 1*C* is dissipated in resistor 1*R* and potentiometer *P* until its voltage is equal to the critical grid voltage of tube *TU*, at which time the tube is rendered conductive. Conduction of the tube energizes relay *TD* and relay *CR* is de-energized. The resistor paralleling *TD* coil is a discharge resistor to prevent chattering of *TD* relay. Timing cycle is changed by adjusting the potentiometer.

Are Employed for Highly Repetitive Cycles

In general, the timing period is determined by the size of the timing capacitor and the amount of resistance in the discharge circuit of the capacitor. The timing adjustment on some relays is made by putting more or less capacity in the capacitor circuit. On other relays the value of the capacitor discharge-resistor is changed. On electronic timing-relays using resistor time delay adjustment, a rheostat can be used and remotely located for convenient timing adjustment.

This type timing relay can have a range of from one cycle to 3 minutes. It is particularly adaptable for highly repetitive operations such as for resistance welding. The accuracy of most electronic timers is 2 per cent or better. In general, it is dependent on the accuracy of the relay used in the circuit. Compensating circuits are sometimes incorporated to min-

Fig. 15—Left—Pneumatic timing relay. Rotation of dial on top of unit adjusts a needle valve controlling timing periods up to $3\frac{1}{2}$ minutes. A filter prevents foreign material from entering the needle valve

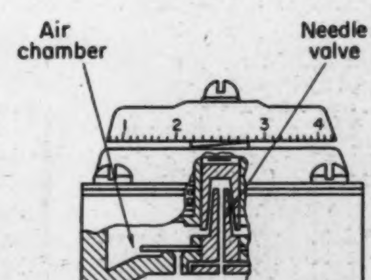


Fig. 16—Right—Sectional view of the timing mechanism for the pneumatic timer

imize error due to regulation of the supply voltage.

An electronic timer can be built using inexpensive components. The reliability of such a timer may, however, be questionable. Most industrial electronic timers use high-grade parts and are reliable in operation. With normal tube and relay maintenance, these timers last indefinitely.

Most electronic timers require a short warm-up period of approximately 10 to 30 seconds before they can be put into operation. This is the time required to heat the electron-tube filament. Some newly developed timing relays make use of a cold-cathode tube which requires no filament current and consequently requires no warm-up time. Most electronic timing relays require a short recycling time. This is the time necessary to charge up the capacitor during the reset period. Some relays, however, are available which feature instantaneous recycling.

DASHPOT: Magnetically operated oil dashpot timing relays are suitable for either alternating or direct current. In this relay a magnetic solenoid operates a plunger, the motion of which is retarded by an oil dashpot. In general, this is simple, fairly dependable in operation, and relatively inexpensive. It is usually employed where reliability is more important than exactness of timing. On some relays, considerable time must be allowed between successive timing periods to allow the piston to return to the bottom of the dashpot. Other relays have built-in check valves which permit the relay to reset almost instantly after the coil on the magnetic solenoid is deenergized.

Practical timing range is from one to approximately 60 seconds. Time adjustment is obtained usually by means of a calibrated washer in the piston or valve disks which may be turned to open or close by-pass holes of various sizes in the bottom of the piston. Since the viscosity of oil varies with the temperature, the timing on this relay changes with temperature. Certain new silicone-dashpot fluids are available, however, to give reasonable accuracy over a temperature range of approximately 100 to minus 25 F.

PNEUMATIC: The pneumatic timing relay depends on the restricted flow or transfer of air through an orifice. On most relays of this type, *Fig.* 15, a fixed volume of air is forced from a chamber through a check-valve mechanism by means of a diaphram. Air is then pulled back into the chamber by the diaphram through a restricted opening. The size of the opening is varied by a needle valve arrangement. The smaller the opening, the longer the time required to refill the chamber with air, the delayed entering of the air into the chamber restricting the motion of the diaphram. This motion is transmitted by a link mechanism to a contact mechanism. Referring to *Fig.* 16 illustrating a typical construction, when the actuating pin, attached to the rubber diaphram, is raised by action of the magnet frame, air in the lower air chamber is forced through the air passage into the upper air chamber. Timing is initiated when the force on the actuating pin is removed. The diaphram then tends to move downward due to the action of a spring and is retarded by the air returning to the lower air chambers through the needle valve.

Time Periods Range to Four Minutes

As the actuating pin moves down, the outer end of the operating lever moves up, thereby actuating the contacts on the snap switch at the proper time. The overall timing period is determined by the position of the valve. Variation in the size of the air passage is obtained by rotating the dial located on the top of the assembly.

The range of adjustment on this relay is from 2/10-second to approximately 4 minutes. Since the motion is always the same but the rate of movement is changed to produce different timing periods, the accuracy always remains the same percentage of the time period, namely approximately ± 10 per cent. Motion of the diaphram and its associated linkage is consistent, operation after operation. The repeat trip position of the contact mechanism usually is not, and therein lies the overall error of the pneumatic

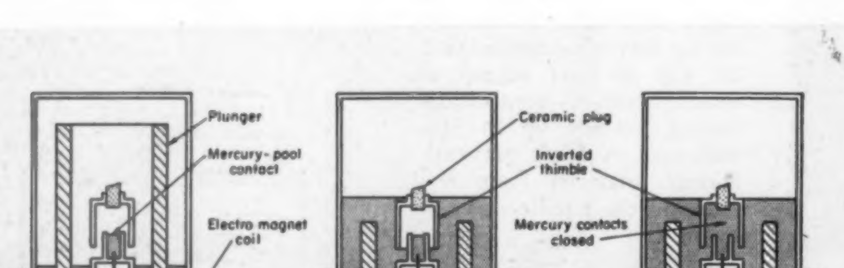


Fig. 17—Mercury time-delay relay. At left, timer is de-energized and circuit is open. In center magnet is energized, displacing the mercury with the plunger and starting the timing period. Mercury is restrained from entering the inverted thimble by trapped gas. This gas leaks through a porous plug admitting the mercury until it completes the electrical circuit with the mercury pool in the thimble as shown at right

relay. It can be seen that if it requires a ⅛-inch movement of the switch mechanism to operate the contacts, a ± 10 per cent timing accuracy could very well be $\pm 1/80$-inch variation in the point at which the contacts on the trip mechanism operate. Actually, however, all this error does not show up in the switch itself. Some error is in the linkages associated with it.

The magnet mechanism which operates against the diaphram system can be mounted on the relay to produce a time delay either after energizing or deenergizing the magnet coil. The relay is practically instantaneous recycling and is usually less expensive than most electronic timing relays.

Most relays are equipped with an air filter so that the relay can be used in relatively dusty atmospheres, without hampering the accuracy of timing. These timers are applied on machine tools, conveyors, and automatic processing equipment where fairly dependable timing is required. Some forms of this pneumatic timing relay are available with either one or two timing steps. The two steps are independently adjustable over the entire range. By using such material as the new silicone type rubbers for the diaphram, practically no deterioration is experienced in the relay. These relays are available with either alternating or direct-current magnets, making them applicable to both types of voltages. On alternating current the contacts are rated for approximately 15 amperes, 110 volts, whereas only a fraction of an ampere is used on direct current. Usually, the contacts are of the snap action variety.

MERCURY: This timing relay embodies hermetically sealed mercury contacts in a metal or glass shell, *Fig.* 17. A magnetic plunger floats in the mercury, and, by means of an electromagnet, is pulled down into the mercury to displace the fluid. The displacing action of the mercury forces some of the inert gas which is trapped in an inverted thimble out through a porus material. Referring to *Fig.* 17, after sufficient gas has been removed from the inverted thimble, the mercury completes a circuit to operate the contact. The timing ranges are available from 2/10-second to approximately 20 minutes. Since these relays are completely sealed, there is no way of changing the time delay of a particular relay. Accuracy is approximately 10 per cent. Usually only one contact is available on the relay, either normally-open or normally-closed. Since the relays are completely sealed against dirt, dust and moisture, and oxidization they are particularly applicable in corrosive atmospheres, and dustry atmospheres. They can be constructed to provide a time delay either on energizing or on de-energizing.

INERTIA: The inertia type timing relay has a special heavy weight mounted on the armature mechanism. The weight is sufficient to introduce the desired time delay in the armature movement after the relay magnet is energized. To eliminate any extra burden on the magnet, the weight is usually balanced. The range of timing is small, namely, approximately 0.08 to 0.1-second. These inertia type relays can be used to make sure that one relay operates its contacts before another when both relay coils are energized simultaneously. Due to its simplicity and low cost, the inertia type is recommended for brief time delays in the order of a few cycles.

Hydraulic Motor-in-Wheel Drives Truck

MANEUVERABILITY is the major requirement of a powered hand truck. The power unit must be readily and smoothly reversible, with steps of power or an infinitely variable multiplication of power, to economically give the performance desired. In designing the Hydro-lift truck, *Fig.* 1, the Clark Equipment Co. developed a package drive incorporating a hydraulic motor within the drive wheel.

An automatic infinitely variable hydraulic pump best met the design requirements from a functional standpoint. A vane type pump manufactured by the Racine Machine and Tool Co. was selected because of its characteristics, price and availability. Belt driven by a 5½-hp, 2600-rpm air-cooled engine, this pump operates at a dead-head pressure of 1300 psi at zero delivery and pumps its maximum delivery of 14 gallons per minute from 0 to 300 psi. Above this pressure, it automatically adjusts its volume as the pressure rises to 1300 psi. At the point of maximum fluid flow, the pump requires just under the maxi-

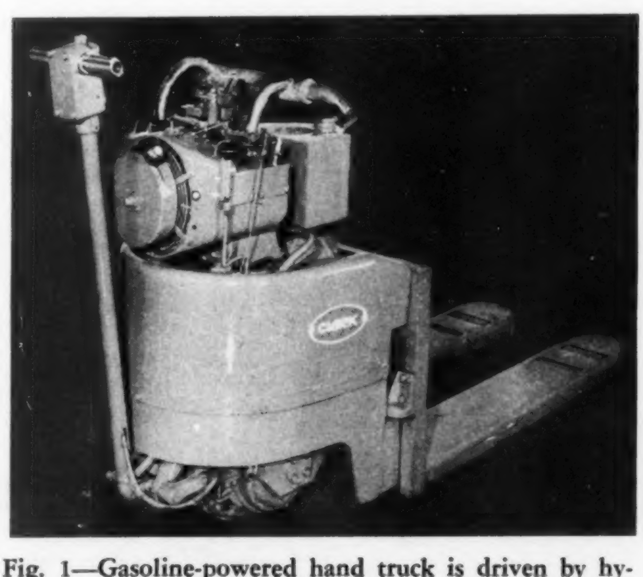


Fig. 1—Gasoline-powered hand truck is driven by hydraulic motor contained in the drive wheel itself

mum output horsepower of the engine. Therefore, the pump cannot stall the engine. It is also possible with this type of pump to select a governor spring which controls the pressure and delivery so that it approximately matches the horsepower curve of the engine. As less horsepower is required at stall to produce the dead-head condition, less heat is generated in the fluid medium, minimizing the problem of heat dissipation.

The wheel-mounted hydraulic motor has a high starting torque and is readily reversible. To accomplish smooth inching and accurate positioning, a valve mechanism, controlled by the engine throttle, feeds low or high pressures into the hydraulic motor. This same mechanism provides for smooth starts when desiring rapid acceleration. Fluid pressure is applied to the motor as a minimum and is raised as the valve closes until pressure becomes great enough to start the machine. The drive pump also supplies power for lifting, lowering and positioning loads.

Capable of developing a large portion of its torque at low speed, the constant-displacement fluid motor is a balanced vane type, manufactured by Dudco Products Co. This motor runs at a maximum of 2300 rpm and will develop at least 90 per cent full torque under 10 rpm.

The motor housing is fixed to one side of a straddle frame, *Fig.* 2. This frame carries the pump and engine assembly and pivots on the main frame for steering. A pinion on the end of the hydraulic motor, *Fig.* 3, extends through a wall within the wheel and drives a triple-reduction gear which in turn drives an internal gear attached to the inner periphery of the wheel hub. The wheel is mounted on antifriction bearings around both ends of the motor frame which serves as the wheel axle. The free end of the motor housing is mounted to the straddle frame by a removable stub shaft. Since lubricant is contained in the end of the wheel which houses the reduction drive, seals are required in the wall between the reduction and the hydraulic motor. Hydraulic fluid to power the motor is transmitted through the end of the straddle frame to which the motor is mounted and directed from there into and out of the motor. Necessity of any flexible hoses is thereby eliminated. A large 14-inch diameter solid tire is pressed onto the exterior of the drive wheel and the external contracting brake operates on the periphery of the wheel.

Permits One-Hand Operation

A mechanical interlock between the throttle and the shift lever forces the operator to return the throttle to idle prior to reversing his direction or shifting to or from neutral. The throttle is co-ordinated with a by-pass valve so that when the throttle is at idle, the by-pass valve is open. Thus it is impossible to shift against a hydraulic load. The machine with the component parts so co-ordinated results in a unit which can be operated with one hand. This includes braking, degree of driving forward, reverse and automatic torque multiplication. The 1200-pound Hydro-lift will carry over four times its own weight up a 12 per cent grade.

Straddle frame mounting of the motor in the wheel allows certain fundamental advantages over existing designs. It makes possible the placing of the wheel closer to the load to make the complete drive more compact, and allows a larger diameter drive wheel to be used.

Gas and electric powered vehicles both have applications where one is more desirable than the other. This motor-in-wheel design is such that the same chassis can be used in the gas-hydraulic machine as is used in the electric machine. Designs are such that even identical reduction gear ratios are used; hence the production of the two "power heads" for both styles of hand trucks is greatly simplified.

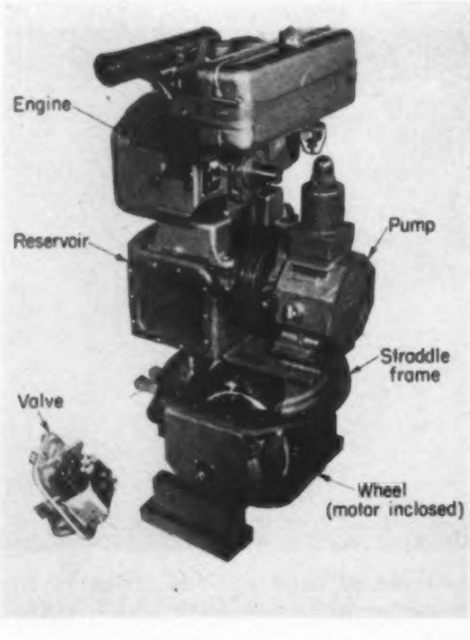
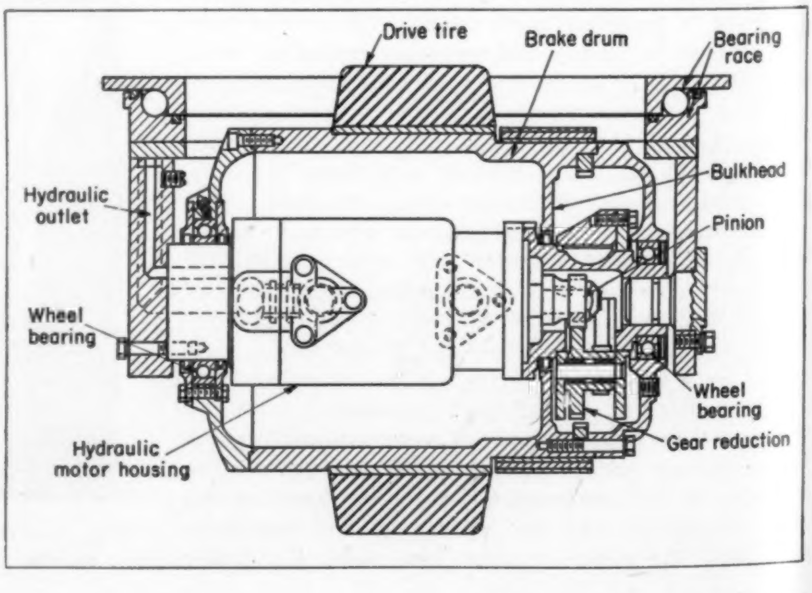


Fig. 2—Left—Power unit is a package assembly
Fig. 3—Below—Construction of power wheel obviates flexible-hose



Push-Pull and Torque Tubes

Important factors in their design including a new knurled-swaged assembly

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COMMON function of all push-pull rod assemblies is the transmission of force and motion from one machine component to another. The rod may carry force in a single direction, change the direction through a bellcrank, or change reciprocation to circular motion as in the driving mechanism of a locomotive. The shapes and sizes, methods of fabrication and types of attachment to adjacent units are as varied as the number of installations. This article surveys the currently used types of push-pull rod and torque tube assemblies and compares their functions and fabrication. It also introduces a recently perfected knurled-swaged method of manufacture which shows superior joint strength over the more commonly used riveted joint in both static test and fatigue life and which can be assembled in less than one-half the time.

In aircraft, many rod assemblies are used for the operation of equipment such as carburetors, canopy locks, selector valves, landing-gear lock mechanisms, etc. In these secondary structures use is intermittent; two to twenty times each flight under relatively light loads. Depending upon their location they are subjected to various vibrations but seldom to vibration under continuous load or rapid reversals of load from tension to compression.

In primary structures for operating the control surfaces of the airplane, rod assemblies are in continuous operation throughout the flight of the aircraft, with loads ranging from very light to very heavy. Rod assemblies used for the operation of bomb-bay or landing-gear doors may be subjected to a minimum number of heavy buffeting loads. Because of the multiplicity of control items in a relatively small area, space is often very critical.

With pressure bulkheads becoming more prevalent, a smooth outer diameter free from projections such as rivet heads or welds is desirable because of sealing problems when operating the rod assembly through a hole in a bulkhead. Especially when used with reciprocating-engine aircraft, vibration is a factor tending to wear clevis pins and loosen jambnuts and rivets. The end fittings must be as small as possible to reduce weight, consistent with the strength

desired. Thus, the dominant factors which influence the design of a rod assembly are:

1. Load—compression, tension, or vibration
2. Length—as it affects column strength
3. Space allotted—therefore diameter
4. End fittings—attachment to adjacent operating units.

SELECTION OF COMPONENTS: A survey of many rod assembly designs in use reveals the following general pattern. The most commonly used alloys for tubes in rod assemblies are 24S-T3 aluminum alloy and SAE 4130 steel, since these alloys offer the best combination of strength, weight and cost. Although others are readily available they entail some sacrifice in strength or weight. The diameter selected may be large because of column strength requirements but this necessitates reduction to fit available rod-end bearings. Wall thicknesses are usually selected from the following group:

1 x .035	¾ x .035	⅝ x .035	½ x .035	⅜ x .035
1 x .049	¾ x .049	⅝ x .049	½ x .049	⅜ x .049
1 x .065	¾ x .058	⅝ x .058	½ x .065	
1 x .095	¾ x .065	⅝ x .065		

The basic types of rod ends attached directly to the tubes are shown in *Fig.* 1. Adjustable components, also called rod ends, are shown in *Fig.* 2. The majority of all rod assemblies use the male threaded fixed rod end in sizes ¼-28, 5/16-24 and ⅜-24 and depend on adjustable components on both ends for attachment to adjacent units. This is due in most part to the fact that a male threaded rod end can be made with larger thread diameter than the female type for a given tube size. However, the female threaded fixed rod end has a distinct advantage over the male end in that the smallest diameter of the assembly may be closer to the point of attachment, *Fig.* 3.

When adjustable units are used on both ends of the assembly it is necessary to "safety" the tube after adjustment to prevent its rotation. This is sometimes done by riveting through the threaded shank on one end or a tang washer is used with a check nut. Although the adjusted length is not changed by rotation there is the possibility that the tube assembly will turn completely off one attachment, *Fig.* 4.

METHODS OF ASSEMBLY: The size of tube and type of rod end selected for a push-pull or torque rod are more or less determined by the load and the attachment to adjacent operating units, but the method of assembly is at the discretion of the designer. He is influenced, to some extent, by the equipment available, but mostly by his concept of what is quickly, easily, and economically put together. The riveted joint has remained the conventional method up to the present time. Although not quickly done it is a positive method, easily inspected. *Fig.* 5 shows various

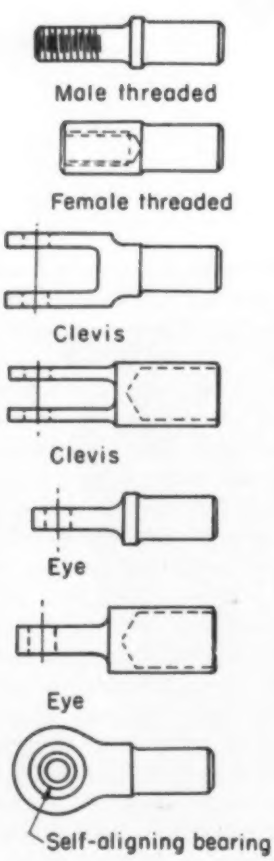
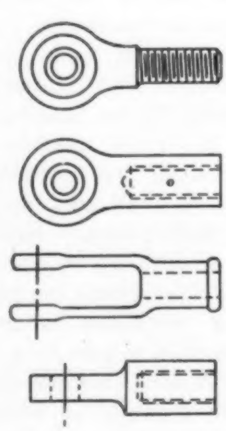


Fig. 1—Left—Some basic types of fixed rod ends

Fig. 2—Below—Commonly used adjustable rod ends

Fig. 3 — Right — Relative position of smallest diameter makes the female fixed end advantageous



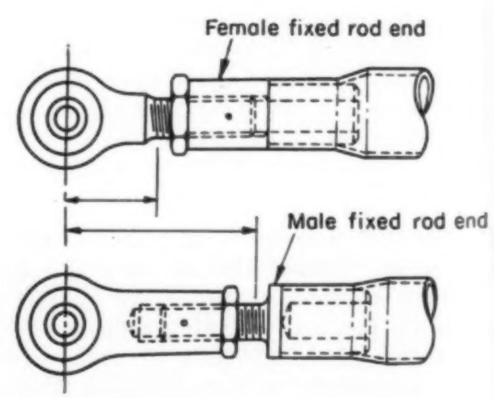
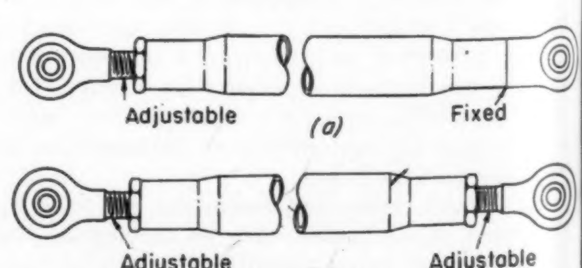


Fig. 4—Below—Adjustment at one end of a push-pull rod (a) is preferred. That at both ends (b) is not recommended



methods of assembly. Their advantages and limitations are discussed in the following:

Tube expanded, rod end riveted in (*a*). This design is preferred for riveted types since it does not reduce wall thickness as much as reaming. Limit of expansion is 5 per cent of the OD of the tube for minimum reduction of cross-sectional area. Tolerances on expanded ID are plus 0.003-inch or minus 0.000-inch. Maximum depth of expansion preferred is 1½ inches.

Tube swaged to controlled ID, rod end riveted (*b*). This design can be swaged to a controlled ID if reduction is not more than 20 per cent of the OD (15 per cent for 24S-T3) and the ID tolerance is 0.005-inch or greater. To permit using standard swaging dies the selected ID should be such that the OD falls between 0 and 0.020-inch above any standard swage die diameter, *Fig.* 6.

Tube reamed, rod end riveted (*c*). This variation allows a close-tolerance ID but reduces wall thickness and leaves a line of stress concentration at the bottom of the reamed hole and therefore is not recommended.

Tube swaged and reamed, rod end riveted (*d*). Diameter reduction should not exceed 15 per cent for 24S-T3 or 30 per cent for SAE 4130. Greater reductions require annealing or normalizing and consequently straightening, an expensive process, *Fig.* 7.

Tube and rod end welded (*e*). This is used for steel alloys only. It requires normalizing, straightening, and finish matching after welding. Cadmium plating and magnetic inspection follow. If assembly is closed at both ends, small holes must be drilled in tube wall, oil filled, and drained, plugged with self-tapping screws.

Tube reamed, expanded or swaged, rod end furnace brazed (*f*). This is used for steel alloys only and is not recommended. It requires less than 0.005-

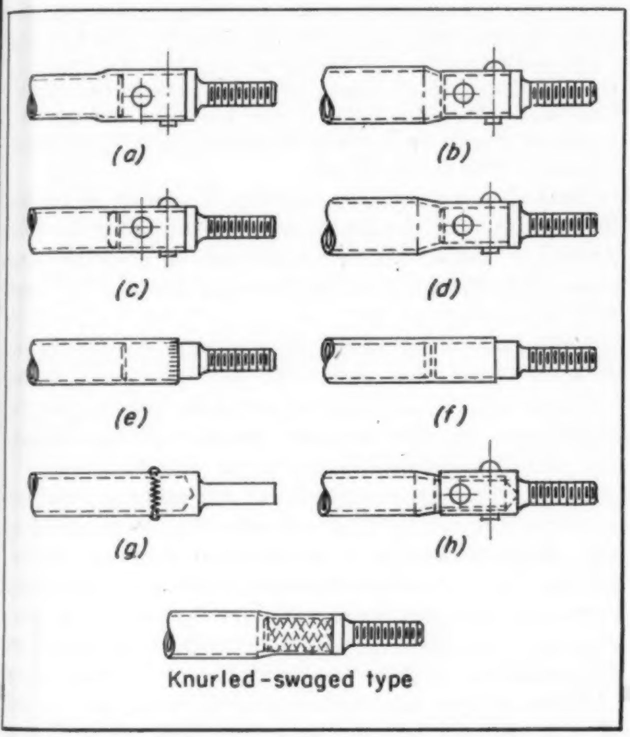
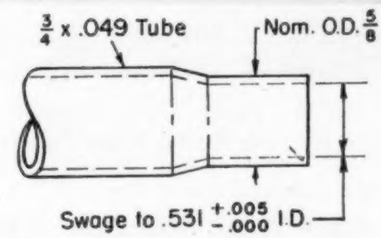


Fig. 5—Above—Methods of assembly currently used for push-pull and torque rods

Fig. 6—Right—Design for swaging to controlled ID



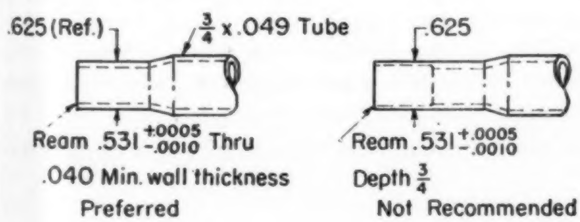


Fig. 7—Preferred design for reaming swaged tubes

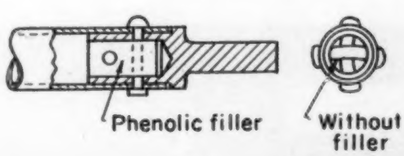


Fig. 8—Above—Filler must be used in hollow rod ends in riveted assemblies to prevent buckling of rivets

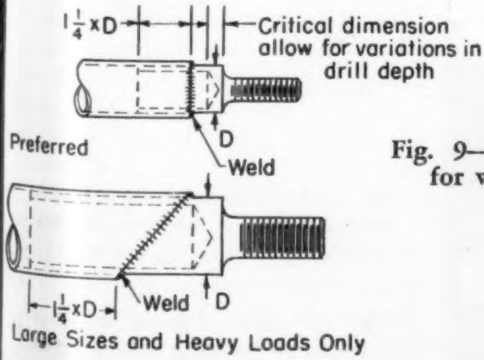


Fig. 9—Left—Preferred designs for welded rod assemblies

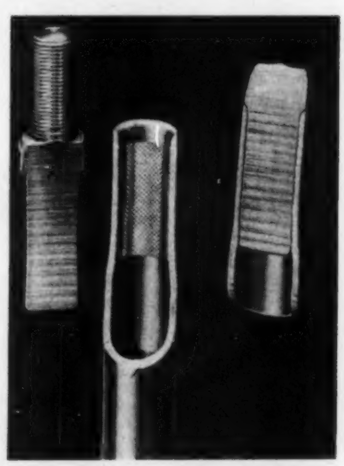


Fig. 10—Right—Knurled serrations become imbedded in the tube wall. End must be harder than the tube for successful results

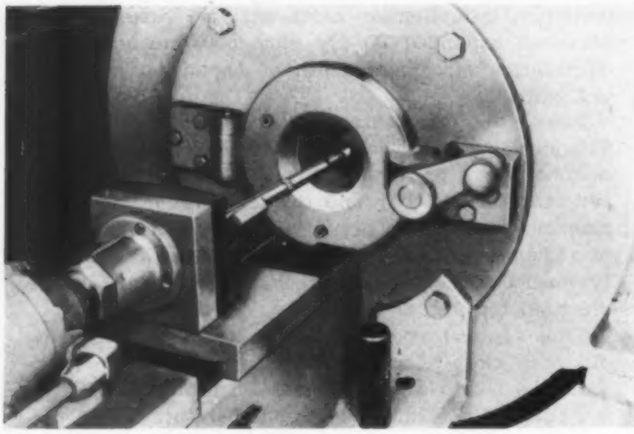


Fig. 11—Above—Rotary swaging machine employed for assembly of knurled-swaged rod

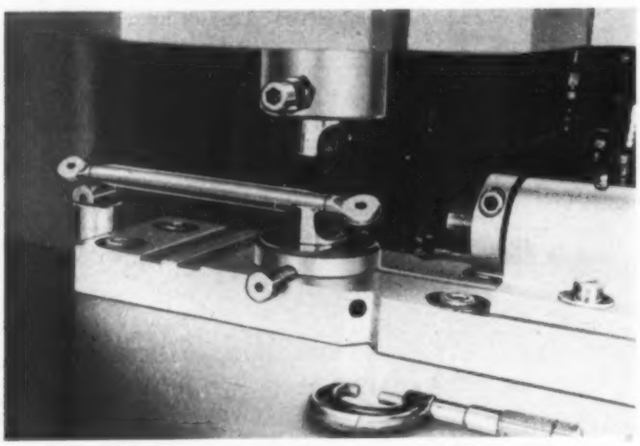
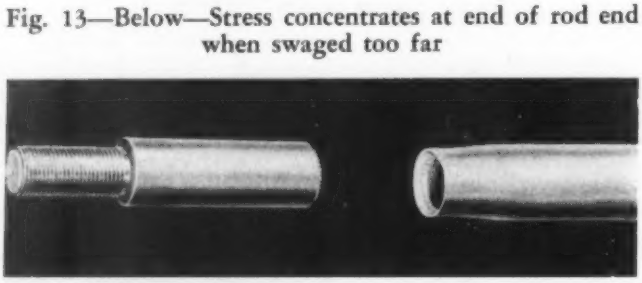


Fig. 12—Above—Punch press operated swaging dies used for rods having large-diameter ends

Fig. 13—Below—Stress concentrates at end of rod end when swaged too far



inch total clearance between tube and rod end. Parts must be carefully cleaned for assembly and tack welded or clamped to hold together while brazing. Allowable load is limited to 10,000 pounds per square inch of area in shear. Processing is the same as for (*e*).

Tube and rod end flash welded (*g*). This design is not recommended. It is uneconomical except in very large quantities and fusion welding is used as an alternate. Applicable to steel alloys only, flash welding requires careful machine setting and weld sample certification. Processing is the same as for (*e*).

Tube swaged, rod end riveted on outside (*h*). This method requires a filler inside the tube. Swage diameter tolerance 0.005-inch total. It is generally heavier than inside fittings.

Swaged and reamed tubes, *Fig.* 7 should be reamed through the full length of swaged diameter to avoid leaving a line of stress concentration, and the minimum wall thickness after reaming should be specified.

Little weight is saved by drilling out the center of a small rod end. A riveted rod assembly requires a filler, usually a machined phenolic bar, to prevent buckling of the rivets during assembly, thus eliminating part of the reduction in weight, *Fig.* 8.

The rod end in a welded rod assembly should project inside the tube approximately $1\frac{1}{4}$ times the rod-end diameter, *Fig.* 9. A perpendicular cut on the end of the tube is recommended. However, structural requirements sometimes make it necessary to scarf the tube end which adds to the cost of manufacture. No expansion or reaming is necessary in most cases.

Some of the methods described, such as welding or brazing, can be used to develop 100 per cent joint efficiency, i.e., a joint strength superior to the tube strengh. But in some cases this entails a sacrifce in extra weight or extreme care in manufacture; extra length of rod end, scarfing for welding, extra area in shear for brazing, etc. A newer knurled-swaged type joint will develop 100 per cent joint efficiency with no more length of rod end than that required for two rivets, and without ream tolerances.

KNURLED-SWAGED ROD ASSEMBLY: The simple con-

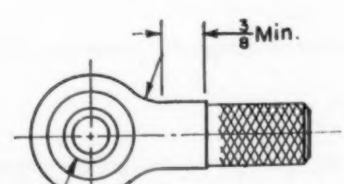


Fig. 14—Above—Knurled rod end bearing. Clearance is necessary for swaging operation

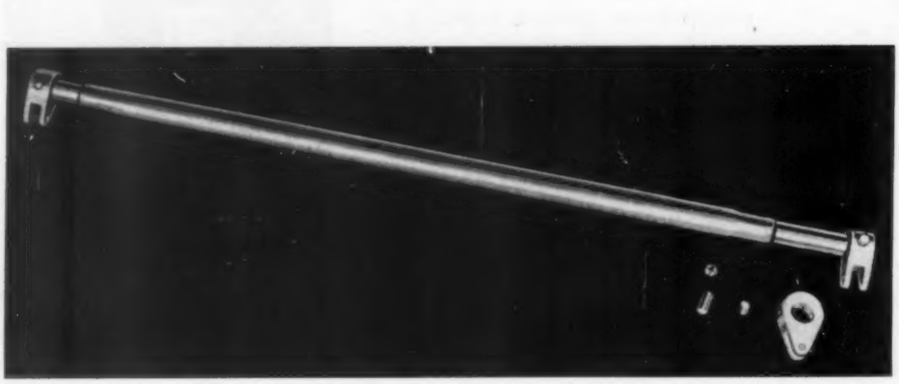


Fig. 15—Left—Torque shaft assembled by swaging onto knurled fittings, then heat treated

ventional rivet method of assembly involves a tube, (cut to length, burred, reamed or expanded or swaged and processed) and a rod end which is inserted and both parts drilled together in a drill jig. Two rivets are installed and riveted by pneumatic gun or squeezer and the drilling and riveting operation is repeated on the other end. If the rod ends are hollow, a phenolic filler must be installed before drilling to prevent buckling of the rivets inside the assembly. This method of assembly requires approximately six minutes to complete.

Assembly Requires Only Two Minutes

By the swaging method, a tube, previously cut to length, burred and processed, is inserted in the swaging machine with the rod end in place. The tube is swaged onto the knurled rod end, *Fig.* 10. The assembly is turned end for end and the operation repeated. This requires less than two minutes including set-up time.

Limitations on Usage: At present, the knurled-swaged rod assembly method is limited to secondary or nonstructural usage. Complete fatigue test and allowable stress data are being submitted for customer approval for primary use. Many parts such as landing gear doors, gun charging access doors and canopy lock mechanism rod assemblies now use the method and will provide further service life information. The knurled-swaged joint has attractive fatigue life and static strength characteristics. It is applicable to both 24S-T3 aluminum alloy and SAE 4130 normalized tubes. The rod ends must be of higher strength than the tube in order to obtain penetration of the serrations into the tube wall, *Fig.* 10, and, therefore, all rod ends of the knurled type must be of 125,000 psi minimum heat treat. The joint is as strong or stronger than the adjoining structure (tube or rod end) in all of the sizes tested, hence the full allowable strength of the tube can be used as the basis for stress calculations. However, the strength of the weakest component of the rod assembly must be considered when establishing the limit load to be applied. This may be the threaded shank or the rod-end bearing. A proof load of 66 per cent of the design ultimate is recommended for the drawing.

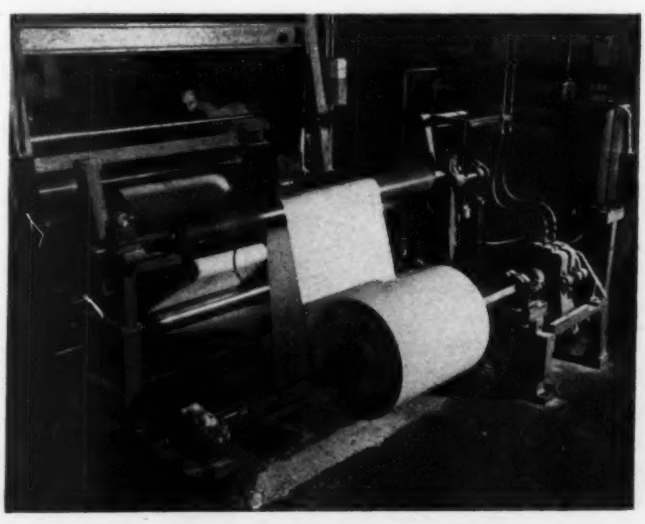
Production Considerations: Swaging is done in either a rotary swaging machine or fast-acting punch press, depending on the size and shape of the rod end. If the entire length of the rod end has cross-sectional dimensions less than the after-swaging diameter of the tube, the assembly can be inserted into the rotary swager, *Fig.* 11. If the rod has a larger diameter, such as the rod end bearing, the assembly must normally be made on the punch press, *Fig.* 12.

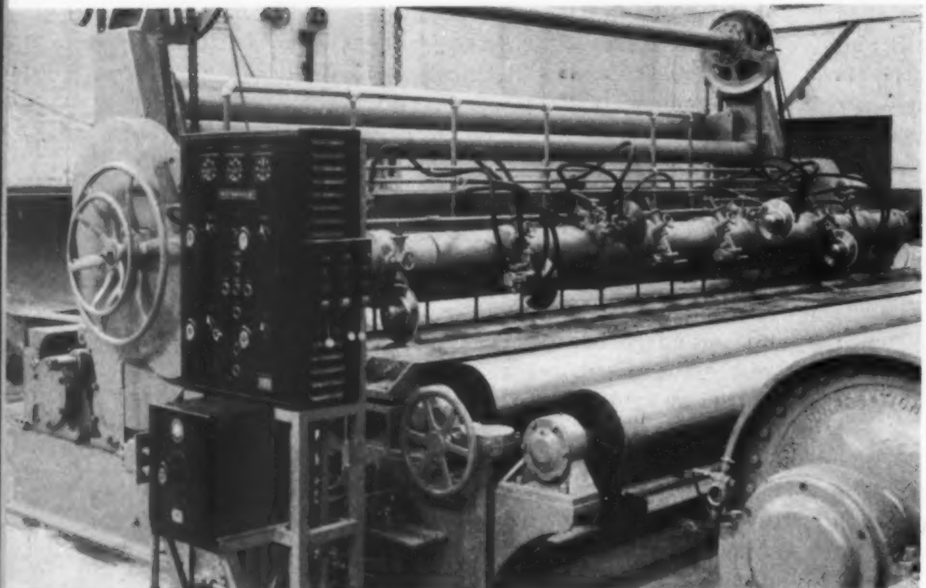
The length of knurled rod end is 1/8-inch greater than the swaged length of tube. This is important because it has been determined that a tube swaged over the full length of the knurl will have a point of stress concentration at the end of the rod end contributing to early failure in fatigue testing, *Fig.* 13. The strength of the swaged joint is obtained through the relative interference between the OD of the knurled rod end and the ID of the tube. This diametral interference nominally ranges from 0.025-inch for an 0.028-inch wall thickness to 0.040-inch for an 0.065-inch wall thickness. In practice the wall thickness of the tube tends to increase as does the length of the swaged portion of the tube and the knurled portion of the rod end. Hence, only two dimensions, aside from the selection of wall thickness, are responsible for producing a dependable joint; the diameter of the knurled portion of the rod end, and the diameter of the tube after swaging onto the rod end. On the standard rod end, *Fig.* 14, the 3/8-inch minimum dimension given in *Fig.* 15 is to permit the operation of the swaging dies which must begin swaging at the end of the tube.

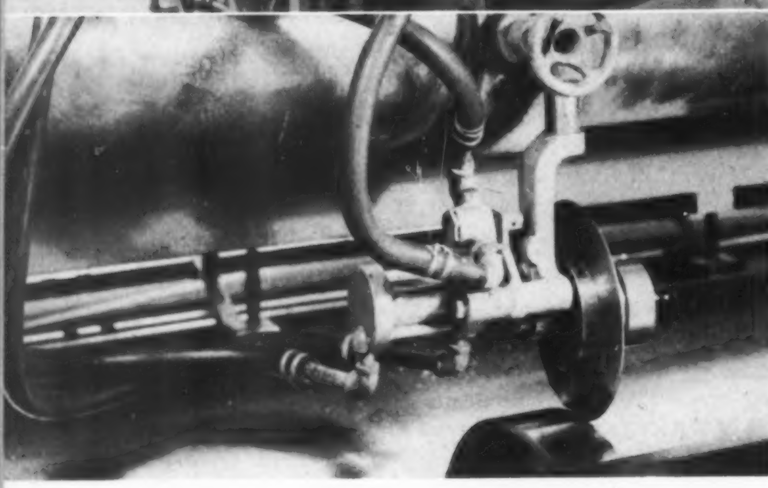
The torque shaft assembly shown in *Fig.* 15 is typical of advanced designs utilizing the knurled-swaged method of assembly. Under static test, the 1-inch diameter x 0.065-inch wall SAE 4130 steel tube, previously heat treated to 160,000 psi, minimum, parted at 32,650 pounds, the joint holding together. A fatigue test at plus and minus 9600 pounds, completely reversed, ran to 61,000 cycles, again failing the tube only. In torsion, the taper pins which held bell cranks in position sheared off at a value of 210 per cent of the design ultimate. These tubes were assembled by swaging and heat treated after assembly.

Controls Edge Electronically

Waxed paper for bread wrappers is slit and rewound in one electronically guided operation on this machine built by the Cameron Machine Co., Brooklyn, N. Y. The electronic scanning head of the Westinghouse slitter regulator can be seen in position in the middle foreground over the moving rider roll. Impulses from the scanning head govern the motor which moves the unwinding and scanner rolls to the right or left as required to keep the sheet edge straight as it unwinds from the roll in the foreground.







Paper an

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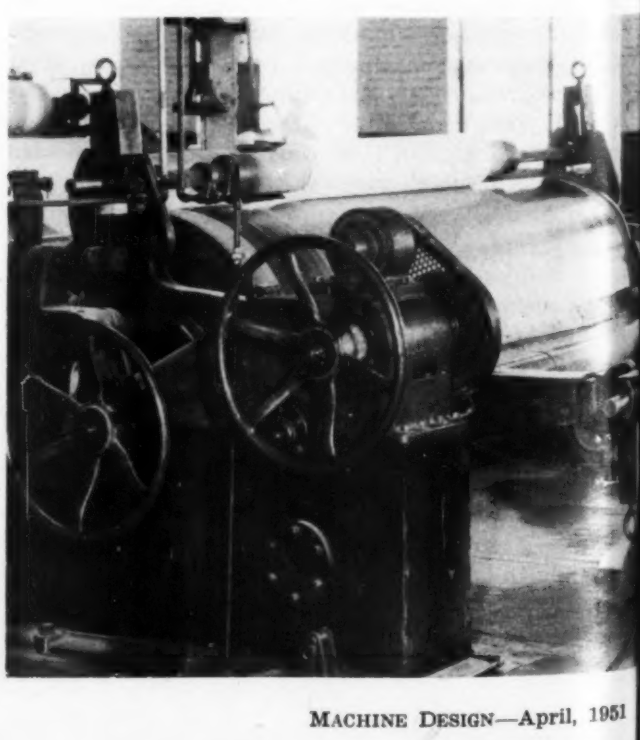
IT MIGHT be said that paper and textile machinery very nearly typifies the overall drive and control problems encountered in general machine design. Certainly it runs the gamut in many phases — speed, synchronization, horsepower, size, control

The rotation of this shaft raises all except the edge slitters out of contact with the sheet. To lower the slitters into contact, the reverse cycle is as follows: the slitters rotate down into position and air is released from each of the small slitters, permitting reengagement. All of this is accomplished by the operator by displacing a single lever-operated air valve attached to the front frame of the machine. This lever-operated valve can be seen directly below the two hand-wheels in the illustration.

Piston Frees and Stops Knives

ILLUSTRATED above is an arrangement* for controlling the upper slitter knife wheels on a paper slitter and winding machine. The circuit is a very unusual one. Air enters the small cylinder shown in the enlarged view to displace the top slitter knife against its spring loading thereby separating it from contact with a lower slitter knife. The end of the piston rod is tipped with Micarta so that it also acts as a brake to stop the slitter from rotation. Machines are usually equipped with several of these slitters and the circuit is so arranged that the air passes from one slitter to the next in series. When the air has passed to the last slitter it is permitted to flow to a master cylinder that rotates the large-diameter shaft on which the top slitters are mounted.

* Patents applied for.



Textile Machinery Drives

...ual drive and control design innovations

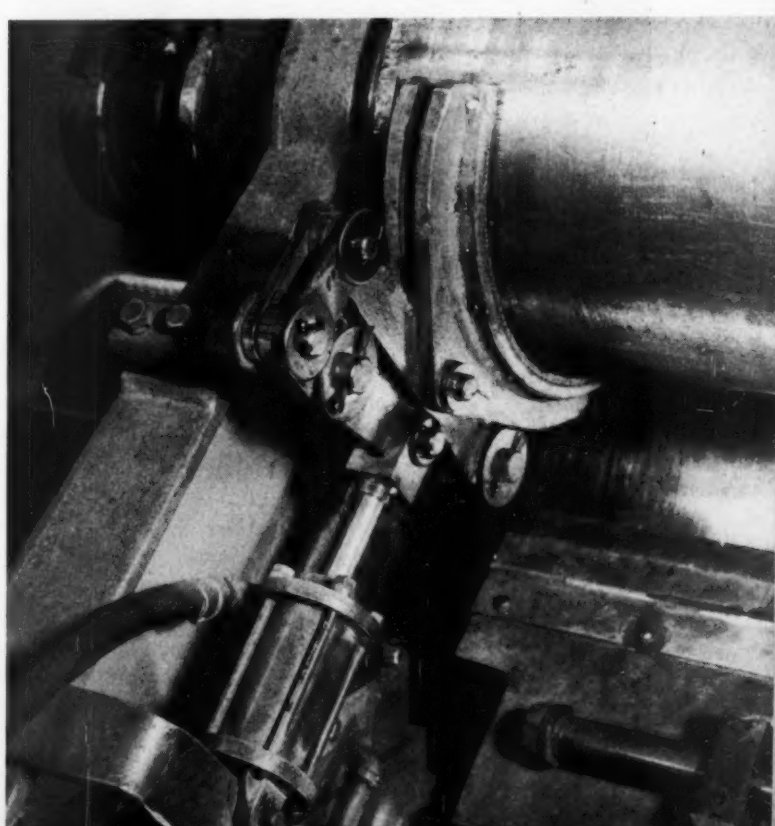
response, etc. To illustrate some of the unusual and novel solutions to vexing design problems encountered in this field, MACHINE DESIGN takes pleasure in presenting the accompanying pictorial article detailing features of paper machinery designed and built by the Rice Barton Corp. of Worcester, Mass.

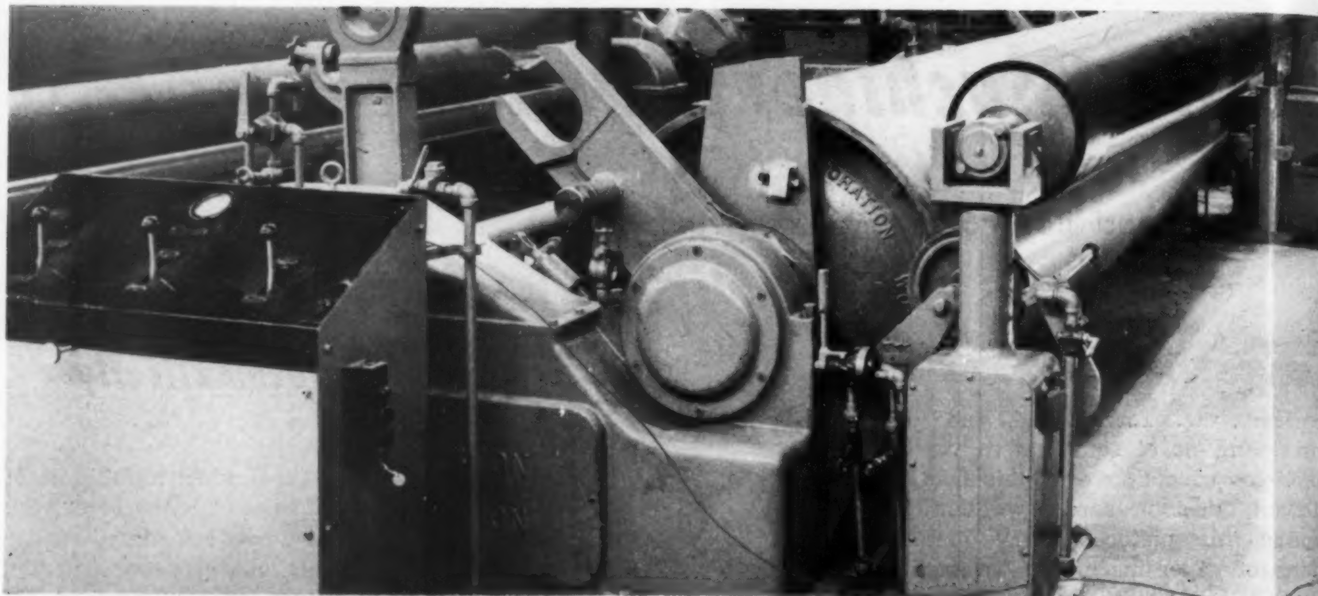
Roll Actuates Centrifugal Brake

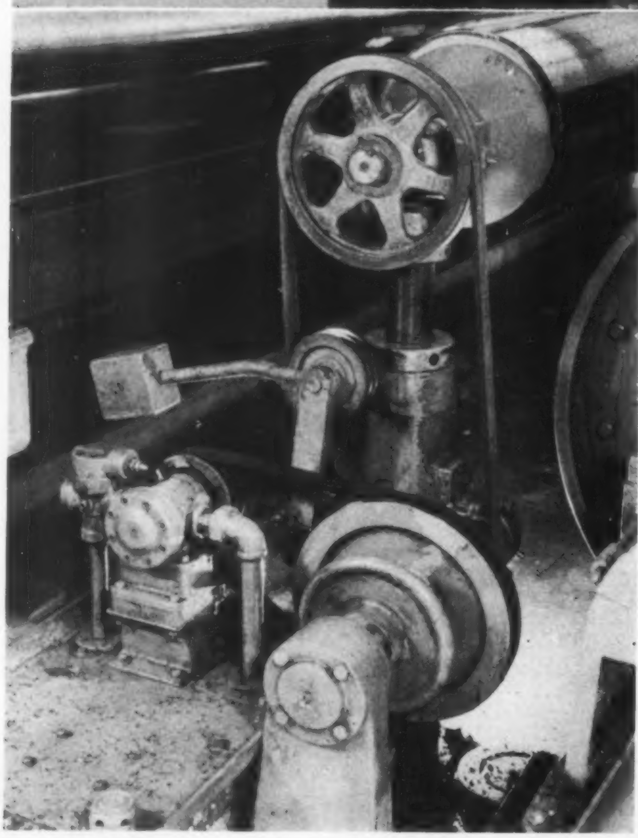
THE device shown below is quite unique. Its purpose is to permit a roll of paper to be lowered from a "12 o'clock" position to a "3 o'clock" position, to a truck or to the floor. Directly above and a little to the right of the handwheel is a centrifugal clutch which is used as a brake. Because the roll tends to fall freely by its own weight and because the speed increases as the moment arm increases, the rotating member in the centrifugal clutch displaces the friction members and therefore provides a brake action which slows and maintains the speed of the roll movement to a predetermined rate. The action is extremely smooth and easy. The empty arms are returned to starting position by means of a handwheel. It will be readily noted that if the arms are to be returned in a matter of a few seconds the centrifugal clutch would also act as a brake in the "up" direction. Therefore, to inactivate this brake, the larger sprocket which drives the clutch has a free-wheeling device which allows the operator to crank the arms up as rapidly as he sees fit.

Stops Rolls Automatically

ON HIGH-SPEED machines, when a full roll of paper is raised out of contact with the driving drum there is a tremendous amount of stored energy in the spinning roll. These rolls of paper on large machines are of large diameter, weigh 18,000 to 20,000 pounds, and rotate at a surface speed of between 1000 and 2000 feet per minute, depending on the grade of paper being made. The WR^2 factor is terrific and, in many cases, if the rotation was not restricted the roll of paper might coast for a matter of ten or fifteen minutes. If this roll of paper is allowed to rotate freely, tension in the outer layers of the sheet is lost and the outside of the roll becomes "loosely wound". It is, therefore, not fit for subsequent operations until all the loose outer layers of paper have been removed from the roll as waste. To overcome this coasting action a brake, which is shown in the illustration below, is used to decelerate the roll within practical limits. The brake is engaged automatically when the operator raises the roll from contact with the driving drum.

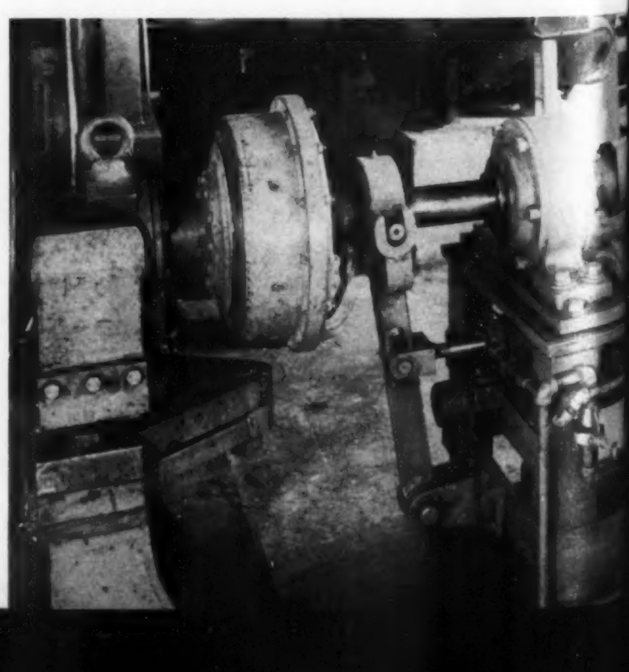


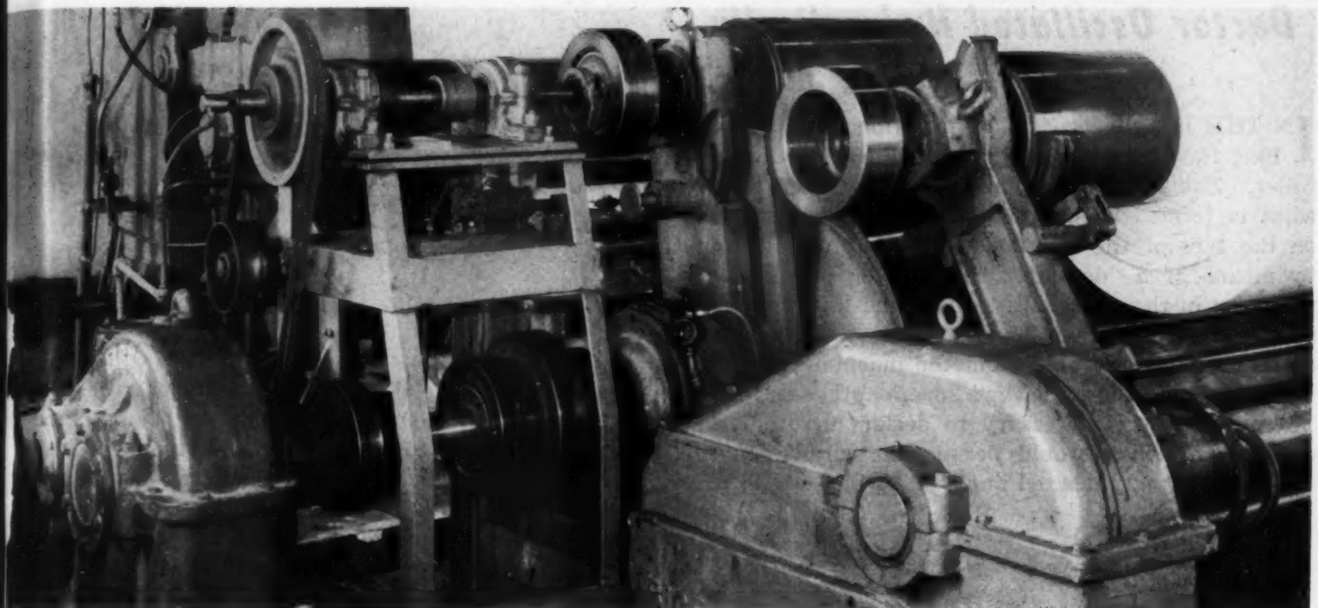




Drag Produced by Hydraulic Pump

THE rubber covered crown roll shown above, is supported on pedestals and is used as a spreader roll on top of paper sheet as it passes to the reel. This spreader roll should produce drag on the top of the sheet, the amount of drag depending entirely on the type of sheet that is to be operated on the reel. In some cases it may be crepe paper and others, flat papers. In order to control the braking effect on this roll a V-belt drive to a Vickers hydraulic vane pump is used with a remotely controlled pressure regulating valve. The sheet of paper which drives the roll also becomes the drive for the pump and as pressure of the regulating valve is increased the amount of work which the sheet has to do is thereby increased. The roll may operate from the same speed as the sheet down to zero because the coefficient of friction of the sheet of paper and the roll would not be sufficient to drive against the pressure setting of the pump. This device works very well because energy is dissipated in the oil and not on friction materials. In friction brakes a means must be provided to overcome and dissipate heat created and adjustment made for expansion of the member as the temperature increases. In this particular case no compensating devices are required.





Air Clutch Accelerates Roll

A ROLL accelerating device* for a paper machine reel and unwinder is shown above. The Rice Barton uniform-speed reel automatically transfers a sheet of paper from one spool to another at speeds of 2000 feet per minute without stopping or slowing down the sheet. In the past when the empty spool has been lowered into contact with the sheet it was necessary for the sheet to accelerate this reel spool from zero to paper speed almost instantaneously. Many times this would cause lightweight sheets of paper to break down or to wrinkle. The device shown was designed to overcome this difficulty. On the end of the spool there is a heavy rim member which has the appearance of a small flywheel. This member of the roll, in the starting position immediately to the left, is engaged by a shaft in the hollow end member. On the end of this shaft is mounted an expanding Fawick Airflex clutch. Air is admitted to this clutch through the rotary joint immediately to the left of the V-belt drive pulley. This Airflex clutch when inflated accelerates the spool to the speed of the machine. As soon as the spool is accelerated, the operator disengages the roll by discharging air from the clutch and retracting the clutch which is mounted on a quill. The retraction of the clutch is done automatically with air by the little cylinder directly under the shaft. The spool, rotating freely at the desired speed, is then lowered into contact with the sheet of paper for the winding operation. The accelerating device remains in synchronism at all paper speeds.

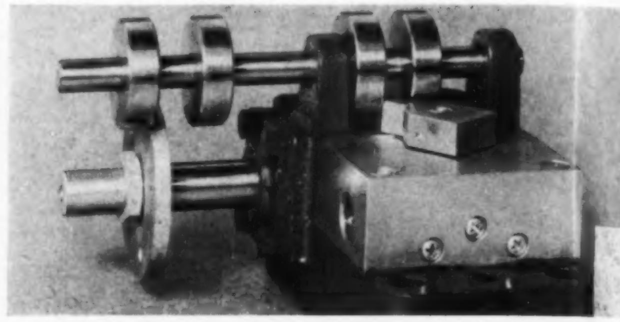
* U. S. Patent 2,475,480.

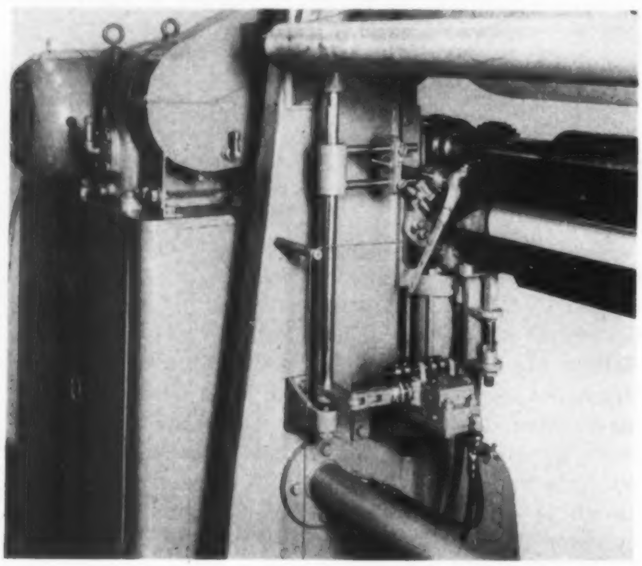
Generator Tensions Sheet

THE device shown in the illustration, left, is a means of connecting a reel spool, with the roll of paper wound thereon, to a generator which acts as a constant-tension braking means as the sheet is being unwound. The connection between the drive shaft and the reel must be able to compensate for angular and lateral displacement of the reel spool so that the sheet may be guided properly into the machine. Located on the quill shaft shown is an Airflex clutch which is internally contracted. This arrangement provides the additional diameter needed for the size of the coupling to transmit power to the generator. This power in many instances approaches or exceeds 150 hp. The quill arrangement here is similar to that of the internally expanded unit in that the external clutch member can be shifted into or out of engagement by air cylinder and clutch throw-out collar designed for this purpose. This design is unique in view of the fact that where low horsepower is required to accelerate the reel spool a small clutch is used internally and where heavy horsepowers are required in a subsequent operation on the same spool a large clutch is used externally.

Doctor Oscillated Hydraulically

IN THE Intaglio process for printing cloth and the like, the pattern to be printed is engraved into a roller. The color is supplied to the engraved roll by what is termed a furnisher roll. The surplus color on the face of the roll is scraped off of the surface by means of a "doctor". This doctor permits only the color in the engraving to pass and be transferred to the cloth. It oscillates from side to side and has to be readily removable from the machine for cleaning, honing of the blade and several other purposes. Therefore, it was necessary to design an oscillating mechanism which would permit the adjustment of the doctor at any desired angle and which would provide for increase or decrease in diameter of the engraved rolls—the minimum diameter being from 14 to 16 inches up to 54 inches circumference. As indicated in the illustration at the left, this is accomplished by means of a vertical keyed shaft which is angularly displaced by means of a hydraulic oscillating unit. On the upper portion of this shaft is a two-prong member. This member is adjustable up and down the vertical shaft to compensate for the change in vertical position of the doctor for various diameter rolls. Two fingers of this member support a horizontal adjustable member which is equipped with a ball and socket bearing between the two fingers. Into this ball and socket member passes the doctor oscillating finger. This ball and socket seat takes care of the combination of movements such as a lateral displacement and any small combination of vertical displacements.





The hydraulic oscillating unit for providing continuous reciprocating motion is shown above. Developed by Rice Barton in conjunction with Vickers Inc., the device has the rate of speed controlled by the rate of flow and the stroke controlled by the distance between the adjustable spools on the valve operating shaft.

Roll is Self-Adjusting

THE center roll shown in the illustration below, is a Ziegler guide roll, a patented device.* It guides a moving web of felt or like material without the use of complicated mechanisms. Most guides are equipped with screws, ratchets, pawls, air cylinders, or similar devices. However, the only moving element of this guide is the roll itself. The roll rotates freely with the travel of the felt and as the felt moves to one side it contacts the stationary extension of the roll. Contact with the strip causes the extension to rotate and the end of the roll travels up the inclined plane, changing the roll axis and guiding the felt back to central position. As soon as the felt is guided out of contact with the extension of the roll, the end then rolls back to its initial starting position by gravity. Either end of the roll is free to be displaced in this manner. Sensitivity of the guide is controlled by the degree of inclination of the tracks which, as can be seen in the illustration, are adjustable to suit conditions.

* U. S. Patent 2,008,318.

A New Approach to Production Winding

Variable-speed drive automatically delivers constant linear speed

By L. A. Graham
Graham Transmissions Inc.
Milwaukee, Wis.

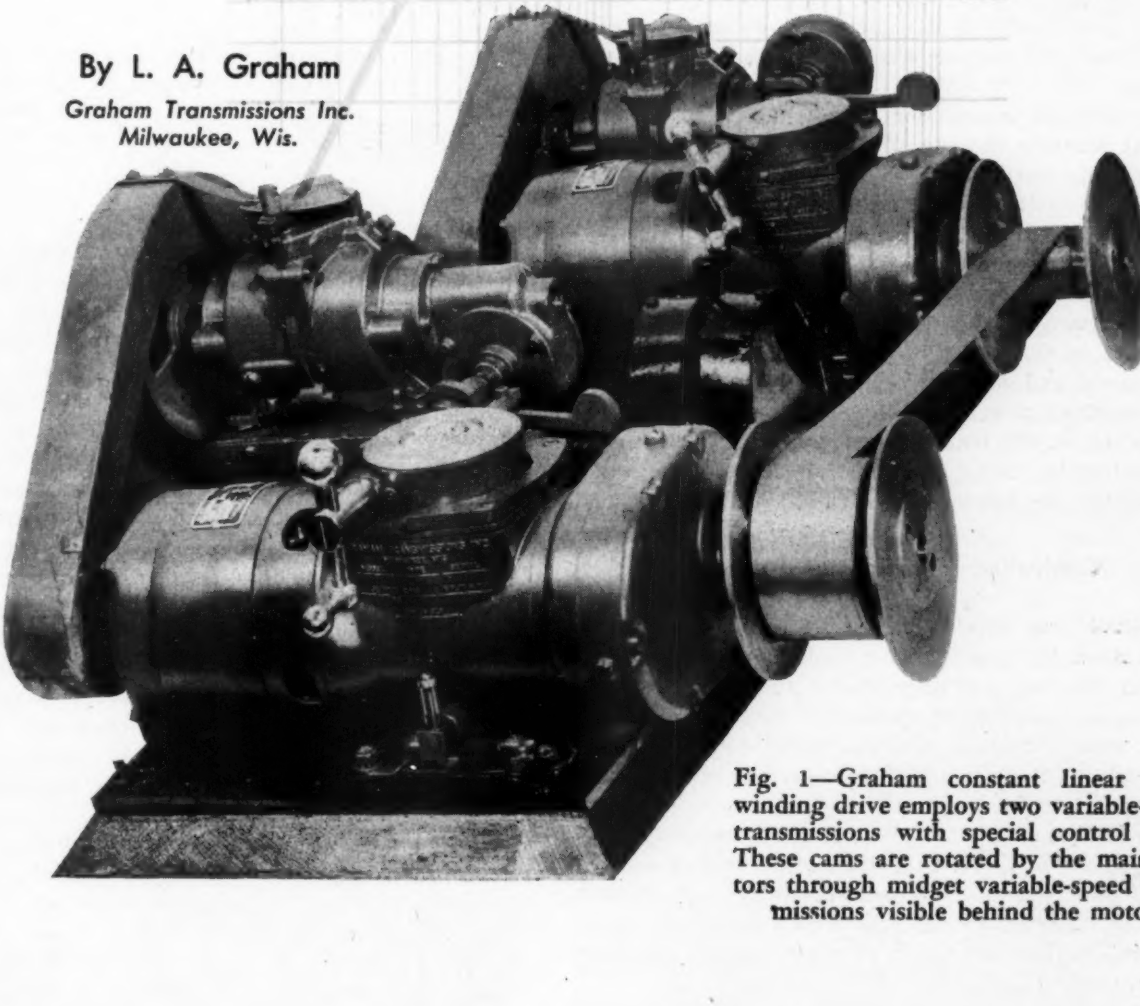


Fig. 1—Graham constant linear speed winding drive employs two variable-speed transmissions with special control cams. These cams are rotated by the main motors through midget variable-speed transmissions visible behind the motors

PROBLEMS of winding and unwinding materials such as cloth, strip, film, paper, and wire are many and varied but the same objective underlies nearly all—a smooth reel with properly controlled tension. Since the material is usually fed to or drawn from the reel at a constant linear speed by capstan, extruder, drawing die, etc., the speed of revolution of the reel on a winding job obviously must start at a maximum and reduce as the winding proceeds, in substantially inverse proportion to the diameter; otherwise the tension could not be maintained at the desired amount.

Until now there has been no attempt to change the speed of the reel in the required relation by merely imposing, on the variable-speed drive, the *calculated* speed wanted at every instant of the winding. This is probably because such a method requires high inherent accuracy in the variable-speed drive. Instead, the variation in the tension itself is commonly used to make the necessary change in the speed of the

reel. This means that the speed continuously has to become wrong and be corrected. For example, in the familiar dancer roll arrangement the winding proceeds at the wrong speed—too great for the job—until the tension builds up to an excessive amount, whereupon the pull of the material overbalances a weight and reduces the speed of the variable drive through an appropriate linkage. The dancer roll continues to move, up and down, until the winding is completed.

Another common method dispenses with the variable-speed drive and uses slipping clutches—mechanical, hydraulic or electric (torque motors)—which slip away the excess speed and power as the winding proceeds. Such clutches, in addition to being inefficient, periodically go out of adjustment and, being usually constant-torque rather than constant-tension devices, leave much to be desired. However, they are in wide use and are satisfactory for many practical purposes.

Other devices in common use include: (1) a linkage that contacts the top of the reel and changes the speed of the variable-speed drive as the reel builds up; (2) electrical contactors that engage the material and are actuated by its excess sag or tautness so as to correct the speed setting; (3) hydraulic or electrical drives that are actuated by the change in wattage drawn by the electrical driving motor. They are based on the fact that a winding job at constant linear speed and constant tension theoretically should draw constant power, therefore an excess in tension is reflected in the wattage and can be used through a suitable pilot motor or similar device to reduce the speed of the reel as required.

Controlled vs Constant Tension

Obviously, no one arrangement can best suit all applications but there is a definite demand in many winding jobs for a simpler, faster and more accurate drive than is presently obtainable, and in particular for a drive that will give controlled tension rather than approximately constant tension or constant torque.

The Graham "constant linear speed transmission," *Fig.* 1, is probably the first attempt to furnish a drive which, once started, will automatically and without recourse to external control, turn a drum on which material is being wound at any desired constant linear speed.

Underlying the design of this packaged drive is the basic idea that the speed of the main variable transmission that drives the reel is to be constantly altered in just the required amount by a cam driven at a constant speed of rotation by another, "midget" variable-speed transmission constituting part of the package; cam speed is set and maintained at the proper figure for each particular job of winding, making the whole process fully automatic, rapid and accurate within limits precise enough for practical needs.

Design of a constant linear speed transmission to operate in connection with a control cam turning at constant speed involves two distinct mathematical calculations: first, derivation of the formula for the shape of the control cam on the main transmission (with the hope that the same cam would then prove applicable to all jobs of winding within the capacity of the transmission); second, derivation of the formula for the speed of the auxiliary midget transmission that drives the cam. Corollary problems, of course, are then to make sure that the cam can be

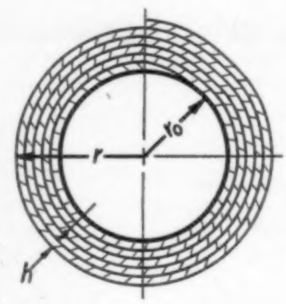


Fig. 2—Length of material wound on reel equals average periphery times number of turns on reel

made commercially and that the variable transmission is of such inherent accuracy as to respond with the required exactness to the cam control.

In the formula for the shape of the control cam on a constant linear speed transmission, the polar coordinate θ is used for the angular movement of the control cam from any assumed starting point and ρ for the cam radius which determines the speed of the transmission. Here the Graham transmission proves especially useful since it already uses a grooved cam for speed control. The standard cam, however, is designed to give an exactly linear relation between angular cam travel and output speed, so that the indicating dial on the transmission may be calibrated to read in revolutions per minute—just like a speedometer—with equal scale divisions corresponding to equal increments of speed. It is necessary, then, merely to find how the shape of the cam has to be altered to do this job of constant-velocity winding.

In the general case the initial radius of the reel at the start of the winding is r_0 inches, the initial reel speed is N_0 rpm, the constant linear speed of winding is F inches per minute, the thickness of the material is h inches, the constant speed of the cam is n rpm, any subsequent reel radius is r inches, the corresponding subsequent reel speed is N rpm, and the corresponding cam angle traversed is θ.

As the winding progresses from the initial radius r_0 to any radius r, the cam having turned through the angle θ degrees, it is clear from *Fig.* 2 that the length wound equals the average periphery times the number of turns:

$$2\pi \frac{r + r_0}{2} \times \frac{r - r_0}{h} = \frac{\pi}{h} (r^2 - r_0^2)$$

which is the familiar formula for the length of a belt or other material wound on a drum. This length must equal the rate of winding F, inches per minute, times the elapsed time t, minutes; also, since the assumed constant speed of the cam for the given condition is n rpm, $t = \theta/360n$, therefore

$$\frac{\pi}{h}(r^2 - r_0^2) = Ft = \frac{F\theta}{360n}$$

Expressing r in terms of N, remembering that $2\pi rN = F$ for all values of r and N,

$$\frac{\pi}{h}\left\{\left(\frac{F}{2\pi N}\right)^2 - \left(\frac{F}{2\pi N_0}\right)^2\right\} = Ft = \frac{F\theta}{360n}$$

or

$$\frac{F}{4\pi h}\left(\frac{1}{N^2} - \frac{1}{N_0{}^2}\right) = \frac{\theta}{360n}$$

which may be written

$$\theta = \left(\frac{1}{N^2} - \frac{1}{N_0{}^2}\right)\frac{90}{\pi}\,\frac{nF}{h} \qquad (1)$$

For any one job of winding, n, F and h by assumption all remain constant, which means that

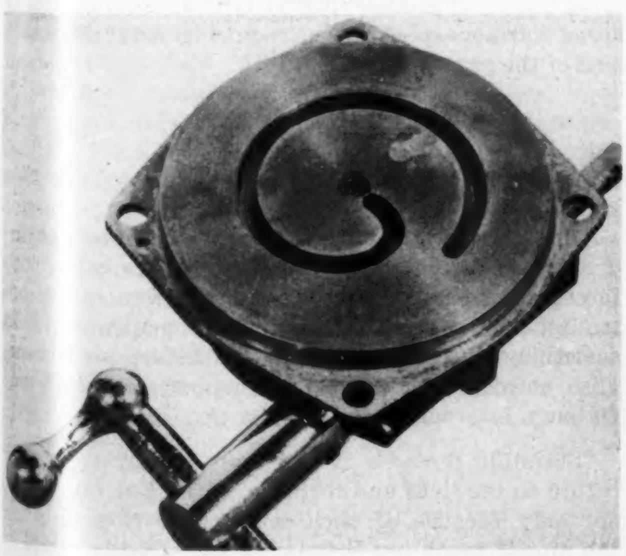
$$\frac{nF}{h} = K \qquad (2)$$

in which K is a constant whose value is to be derived. Equation 1 may then be written

$$\theta = \frac{90}{\pi}\left(\frac{1}{N^2} - \frac{1}{N_0{}^2}\right)K \qquad (3)$$

This equation clearly establishes the shape of the cam (relation between θ and N for all values) provided the desired limiting values of θ and N are set up for any one transmission.

Fig. 3—This special cam replaces standard speed-control cam in Graham transmission. When driven at constant speed it gives constant linear speed to the material while it is being wound



In the drive shown in *Fig.* 1 the desired maximum and minimum values of the output speed were taken as 200 rpm and 40 rpm (this gives an available build-up of 5 to 1 in the diameter of the reel); it was also decided to obtain this speed variation over a cam travel of 360 degrees so as to give a full circle dial for speed adjustment. This at once establishes the relations $N_0 = 200$ rpm when $\theta = 0$, and $N = 40$ when $\theta = 360$ degrees. Substituting these values in Equation 3 gives

$$360 = \frac{90}{\pi}\left(\frac{1}{1600} - \frac{1}{40{,}000}\right)K$$

from which

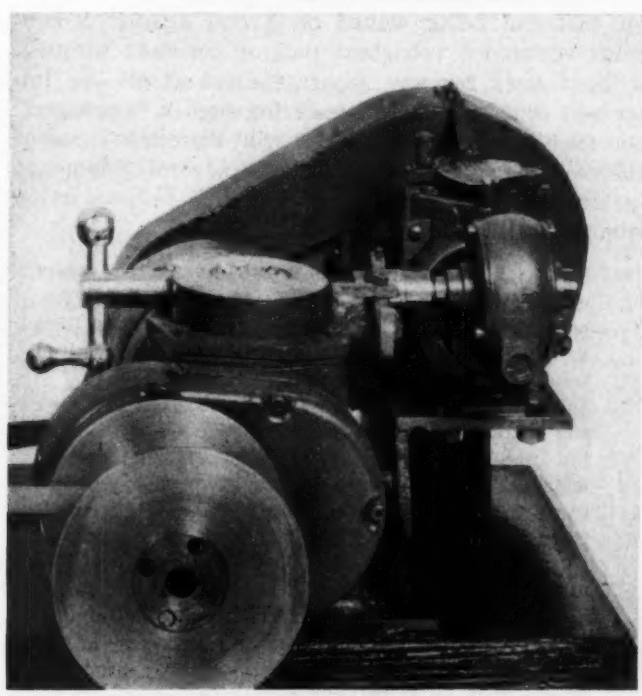
$$K = \frac{360\pi}{90} \times \frac{40{,}000}{24} = \frac{20{,}000}{3}\pi \qquad (4)$$

and the equation for the cam is

$$\theta = 600{,}000\left(\frac{1}{N^2} - \frac{1}{40{,}000}\right) \qquad (5)$$

Equation 5 which shows that θ varies inversely with the square of the output speed, is readily transformed into a curve for the cam shape by substituting for the various values of N the corresponding values of the cam radius ρ. For the drive in *Fig.* 1, this gave the cam shown in *Fig.* 3. It is to be noted

Fig. 4—Drive "package" consists of main variable-speed transmission controlled by special cam driven by midget transmission at upper right. This is another view of the rear unit shown in Fig. 1



that this shape can be readily formed by usual manufacturing processes. Incidentally, the reversing style Graham (in which the output shaft turns in the opposite direction from the input shaft) was used since it results in a better cam shape.

Since the only variables in the equation for the cam shape are the speed and travel, the same cam is obviously usable for all possible variations in reel diameter, material thickness and linear speed that fall within the scope (speed and torque capacity) of the given drive, the only requirement being that the constant cam speed conform to the relation given in Equation 2. This relation establishes at once the formula for the cam speed for any thickness of material and linear speed of winding

$$n = \frac{Kh}{F} \qquad (6)$$

For the example worked out in the foregoing the relation is

$$n = \frac{20{,}000\,\pi h}{3F} \qquad (7)$$

This gives the surprisingly simple relation that the cam speed varies directly with the thickness of the material and inversely with the speed of winding—a formula that may readily be applied in the field for each new job of production winding.

Drive Can Establish Own Speed

So far in the discussion it has been supposed that the constant linear speed drive was required to match a linear speed derived from some other power source. However, there are many cases where the drive here discussed can itself establish a desired linear speed for material being wound on a reel against a constant resistance (constant pull or constant torque), without need for any separate drive at all—an important economy. Many uses for such a "packaged" constant linear speed drive will doubtless present themselves. In this connection, several important incidental features may prove useful in production winding:

1. Although the *initial* speed setting for the start of the winding or unwinding should theoretically correspond to the linear speed to be matched, this can be "shaded" up or down to give a desired tension change as the winding proceeds
2. Inherent characteristics of a transmission in which the output speed automatically reduces by a slight but fixed amount with increment of load can be used to decided advantage for a specific problem in repetitive production winding where the conditions remain substantially constant from one reel to the next
3. Calculated speed of the cam for a given linear speed and material thickness can also be shaded if desired to give a build-up or slackening of the tension as the winding or unwinding proceeds, where such is desirable for specific cases
4. Where the linear speed is presently limited by the rapidity of operation of the external corrective devices, such as dancer rolls, etc., this new technique may not only give a better reel but permit faster operation, thereby reducing the cost of winding.

Two Graham transmissions with built-in 1/6-hp, 3600-rpm motors and giving speeds from 200 rpm down to 40 rpm, are shown in *Figs.* 1 and 4, the drives being equipped with the special constant linear velocity cams just described. The cam on each drive is connected through a simple form of jaw coupling to the output shaft of a midget type Graham transmission that is run at a constant speed of 600 rpm from the same motor that drives the main transmission. Output speed of the midget varies from 7 rpm to zero. Since the cam shafts on the larger transmissions are driven through a 40:1 worm reduction, the cam speed is adjustable from about 1/6-rpm down to zero, which covers the requirements as will be shown.

The two transmissions, in the demonstration unit shown in the photograph, alternately wind and unwind ribbon from one reel to another. Since the diameter of the empty reel is 1½ inches, a top speed of 200 rpm corresponds to about 900 inches per min. With ribbon thickness 0.004-inch and the constant, K, of the cam equal to 20,000 π/3 (Equation 4), the cam speed for this particular linear speed and material thickness is (20,000 π/3) (0.004)/900 or 0.093-rpm, the output speed of the midget being 40 times that amount or 3.72 rpm. In practice the initial winding speed is made slightly above the required amount and the unwinding speed slightly below the corresponding figure to give the desired tension. When the winding is completed each cam is disconnected from its drive by the jaw coupling and rapidly wound back by hand to the proper initial setting.

A cam such as the one whose mode of calculation has been here derived can, of course, be introduced externally to any type of variable speed drive by inserting the desired end values in Equation 3 and obtaining Equation 5. The formula for cam speed for any given set of winding conditions remains as derived above, of course, for any type of constant linear speed variable-speed drive, requiring only the insertion of the proper value of K.

They Say...

"One thing we're too prone to do in American industry is call conferences. We say, 'Let's call a conference and talk it over.' Unless conferences are organized to the point that people are motivated to do something different than they did before, we accomplish nothing."—E. H. Reed, *manager, education and training, International Harvester Co.*

"Scientific research is tied more tightly than ever before to the lives and fortunes of the general public, not only because of such military developments as the atomic bomb but because of the steadily increasing flow of new ideas and products from the nation's laboratories."—Charles E. Wilson.

Designing a Vane Pump

. . . for high-viscosity service

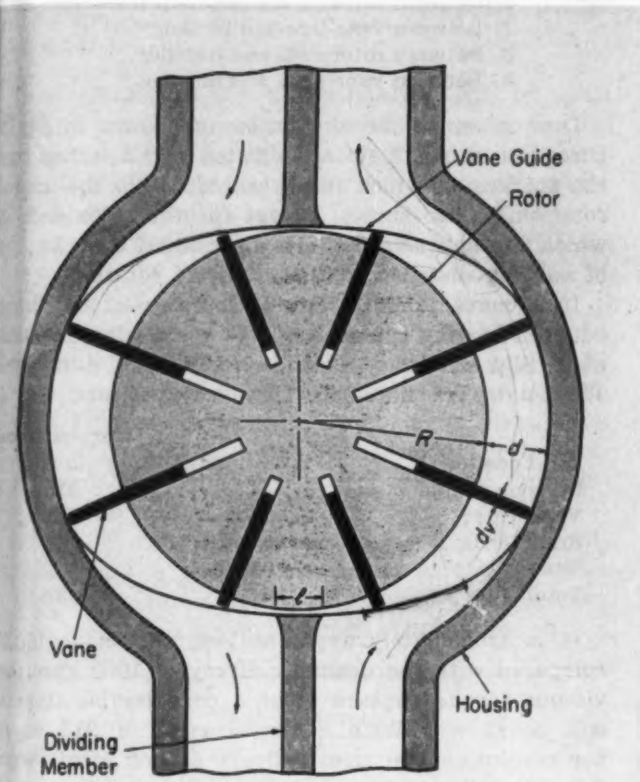


Fig. 1—Simplified section through double-acting vane pump. Clearances can be selected for maximum efficiency with particular viscosity ranges

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POSITIVE displacement pumps are used generally to pump liquids having viscosities that lie in a rather limited range. It is true that, at one extreme, gasoline is being pumped successfully with multipiston pumps. At the other extreme, Bunker-C oil has been pumped successfully for some time. However, it would be possible to pump liquids of extremely high viscosities even more efficiently if clearances were designed specifically for the particular service intended.

The theory of clearance design, set forth previously[1, 2, 3] and not repeated here, will be applied to the design of a pump for a specified unusual service. For this example, a positive displacement pump of the vane type will be studied. The design obtained by this rational method will be compared with a conventional design to determine the efficiencies of the two units under comparable operating conditions.

The basic concept involved in this rational design of clearances is that the performance of a positive displacement pump is governed by the following two factors:

1. Slip, which detracts from delivery of the pump
2. Viscous drag on the moving parts, which makes it necessary to apply a greater torque to rotate the shaft than is required to deliver the liquid against the imposed pressure differential alone.

Since slip increases with the cube of clearance and viscous drag increases inversely with clearance, there is an optimum value of clearance which will result in a minimum loss of power. If clearance is designed to fulfull this requirement under the specified operating conditions, best efficiency can be obtained over a wide range of operating conditions.

The design problem to be considered here is that of determining the proper clearances for a double-acting vane pump to operate under the following conditions: delivery = 1000 gpm, pressure difference = 100 psi and viscosity = 100,000 SSU.

So far as the writer is aware, there is no rotary positive displacement pump commercially available which will operate as efficiently as might be desired and perform the indicated service. The following discussion will point the way toward the design of such a pump by establishing the proper clearances to

[1] References are tabulated at end of article.

be used. The further details of design, such as bearing types and sizes, strength and proportions of various parts, etc., will be omitted since they are routine matters readily handled by any competent designer.

First step in the design of the pump is the determination of the required displacement. In order to establish this element it is necessary to select the operating speed. Ordinarily one would select a low speed for pumping a liquid of such high viscosity (100,000 SSU), but 1200 rpm will be used to reduce the size of pump required. In order to discharge 1000 gpm at 1200 rpm a displacement of 192 cu in. per revolution is required. Since a double-acting pump is to be used, it is necessary to displace only one-half the displacement volume in a single revolution of the pumping element. Required vane area can then be obtained by making a few further arbitrary decisions concerning ratios of lengths. It will be established that (see Nomenclature) $R = L/2$, $d = R/3$, $l = R/4$, and $d_v = 0.15d$.

These elements of the pump are indicated in *Fig.* 1. Displacement volume, D, is related to these dimensions as follows:

$$D = 4Ld\pi(R + \tfrac{1}{2}d)$$

Upon substituting the length ratios and solving for the rotor radius, $R = 2.7$ inch. Corresponding values in inches of the related elements of the pump are: $L = 5.4$, $d = 0.900$, $l = 0.675$, and $d_v = 0.135$. To compensate for slip, letting the design value of $L = 6.00$ rather than 5.4 inches, the total displacement D is 213 cu in.

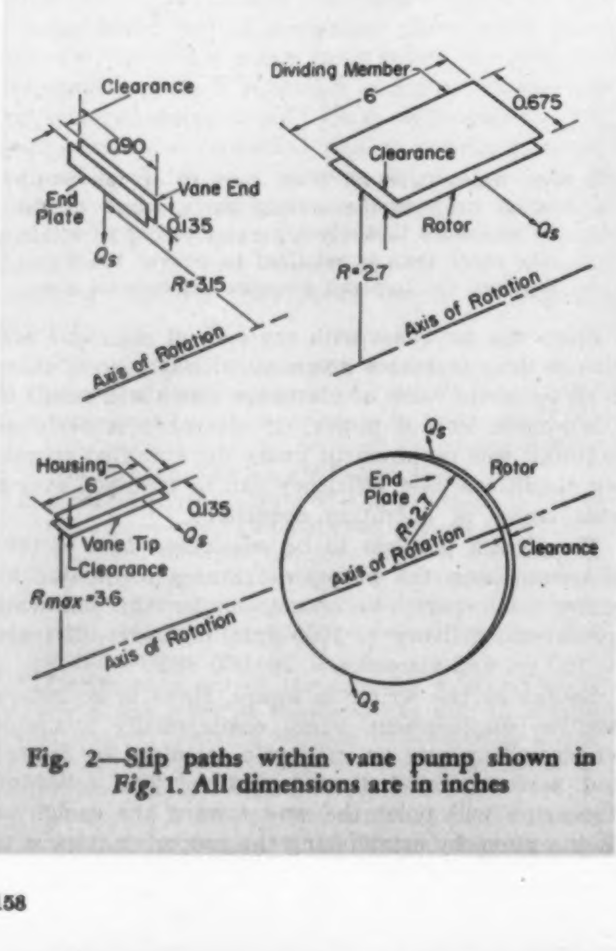

Fig. 2—Slip paths within vane pump shown in *Fig.* 1. All dimensions are in inches

Slip and viscous drag in this pump originate at the following points:

1. Between dividing member and rotor
2. Between vane tips and housing
3. Between rotor ends and housing
4. Between vane ends and housing.

Dimensions of the slip paths are shown in *Fig.* 2. Direction of slip flow is indicated and distance from the surface on which the shear occurs to the axis of rotation is also shown, except for the rotor ends on which the shearing force is distributed over an area of varying distance from the axis of rotation.

In a conventional design a clearance of 0.005-inch between moving surfaces would be considered reasonable. Slip and viscous torque arising at the indicated slip points, calculated for this clearance, are:

Location	Slip (gpm)	Viscous Torque (lb-ft)
Division members	0.019	312
Vane tips	0.048	352
Rotor ends	0.003	868
Vane ends	0.014	51
Total	0.084	1583

It is immediately apparent that slip is negligible compared with the desired delivery of 1000 gpm, but viscous torque appears to be a considerable item as will be shown. With a displacement of 213 cu in. per revolution, the ideal delivery Q_i and ideal torque T_i are as follows:

$$Q_i = \frac{Dn}{231} = \frac{(213)\ (1200)}{231} = 1105 \text{ gpm}$$

$$T_i = \frac{(\Delta p)D}{24\pi} = \frac{(1000)\ (213)}{24\pi} = 2830 \text{ lb-ft}$$

Efficiency of the pump is defined in terms of the volumetric, torque and overall efficiencies which are obtained as follows:

$$\eta_v = \frac{Q_i - Q_s}{Q_i} = 100 \text{ per cent}$$

$$\eta_t = \frac{T_i}{T_i + T_d} = 64.1 \text{ per cent}$$

$$\eta = \eta_v\eta_t = \frac{Q(\Delta p)}{2\pi Tn} = 64.1 \text{ per cent}$$

It must be realized that a pump could not actually attain this efficiency since bearing friction and all other sources of friction except viscous drag on the rotating elements have been neglected. The effect of these neglected sources of friction would be to reduce the torque and overall efficiencies by 5 to 10 per cent.

In order to determine best clearances for the various elements of the pump, losses in power due to slip and viscous drag are added and the clearance that makes the total power loss for each element a minimum is then determined by minimizing the equation for power

loss in terms of clearance. The results of this procedure are as follows:

Location	Clearance (in.)
Division members	0.0340
Vane tips	0.0278
Rotor ends	0.0680
Vane ends	0.0234

These clearances are extraordinarily large by usual standards, but are entirely consistent with the requirements of the pumping service under consideration. The calculated elements of the performance with these clearances are as follows: slip $Q_s = 24.2$ gpm, viscous torque $T_d = 183.8$ lb-ft, volumetric efficiency $\eta_v = 98$ per cent, torque efficiency $\eta_t = 94$ per cent, and overall efficiency $\eta = 92$ per cent.

It is immediately apparent that a sacrifice of two per cent in volumetric efficiency yields a gain of 30 per cent in torque and 28 per cent in overall efficiency. This is a considerable factor in an installation of this large size, since the total ideal power requirement is 646 hp and the saving resulting from increased clearance is 305 hp.

It can be shown[4] that maximum efficiency is obtainable with a given pump at a particular value of the dimensionless parameter, $\mu_n/\Delta p$. The efficiency will be less than maximum for all other values of this dimensionless number. Thus far, performance has been considered at a definite value of this quantity defined by the specified pressure, speed and viscosity requirements. Since it is usually desirable to plan on the pumping of liquids of different viscosities, it is well to consider the effect of viscosity on pump performance.

The two pumps considered previously, namely the one with optimum clearances for service with 100,000 SSU oil, the one with 0.005-inch clearances throughout and, in addition, one with 0.001-inch clearances throughout are re-evaluated, by computation of the efficiencies for operation at 1000 psi and 1200 rpm with oils of various viscosities. Efficiencies obtained under these conditions are shown in the semilogarithmic plotting of *Fig.* 3. The pump with smallest clearance operates with high efficiency when the viscosity of the oil is low; in fact, for all viscosities less than 1000 SSU it outperforms the other two pumps. However, if the viscosity exceeds 10,000 SSU, efficiency of the pump is less than that of either of the other two. At the design viscosity of this problem, 100,000 SSU, efficiency of the pump with optimum clearance exceeds that of the other two by material margins.

It is apparent that each pump is particularly well-suited for operation in a particular range of viscosity. For example, efficiency in excess of 90 per cent can be obtained with each pump in the following indicated ranges of viscosity:

Clearance (in.)	Viscosity Range (SSU)
Optimum	20,000 to 130,000
0.005	100 to 20,000
0.001	100 to 4,000

The pumps with largest and smallest clearances have mutually exclusive ranges of high-efficiency performance while that of the pump with intermediate clearance overlaps the range of the pump with smallest clearance and approaches that of the pump with optimum clearance.

As has been noted earlier, efficiencies indicated in *Fig.* 3—and those computed previously—cannot be realized since bearing and sliding vane friction have been neglected. Such friction may be accounted for by the factor $C_f = 0.05$ in the torque equation:[3]

$$T = \frac{D(\Delta p)}{2\pi} + C_d D \mu n + \frac{C_f D\,(\Delta p)}{2\pi}$$

With this modification, efficiency of the pump with optimum clearances will follow the curve labeled $C_f = 0.05$ in *Fig.* 3. This efficiency curve is similar in general nature to that of this pump when bearing

Nomenclature

C_d = Coefficient of viscous drag
C_f = Coefficient of dry friction
d = Exposed vane length, in.
d_v = Total displacement, cu in.
D = Width of dividing member, in.
l = Vane thickness, in.
L = Rotor length, in.
n = Speed, rpm
Δp = Difference in pressure between outlet and inlet, psi
Q_i = Ideal delivery, gpm
Q_s = Slip flow, gpm
R = Rotor radius, in.
T = Actual torque, lb-ft
T_d = Viscous torque, lb-ft
T_i = Ideal torque, lb-ft
η = Overall efficiency
η_t = Torque efficiency
η_v = Volumetric efficiency
μ = Dynamic viscosity of liquid, SSU

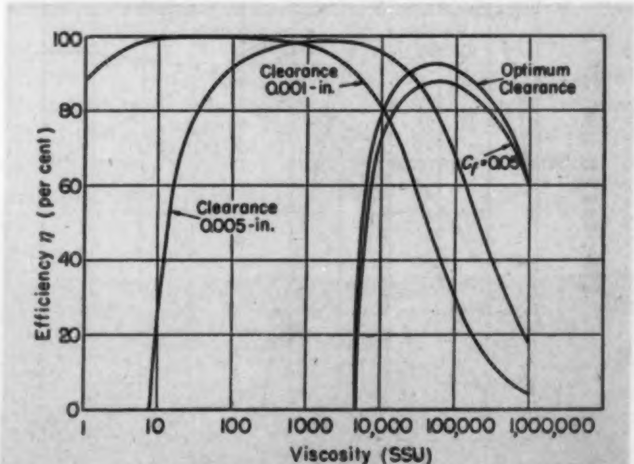

Fig. 3—Total theoretical efficiency versus viscosity of pumped liquid for different clearances. Optimum curve, and its companion with friction included, are based on viscosity of 100,000 SSU

friction is not considered but lies below the first curve consistently, as would be expected.

The effect of clearance on pump performance is indicated in a slightly different manner in the curves of *Fig.* 4. Here efficiency is shown as a function of clearance for three different liquids. One liquid has a viscosity of 10,000 SSU, the second a viscosity of 100,000 SSU, and the third a viscosity of 1,000,000 SSU. It is apparent that a pump of the overall dimensions selected at the beginning of this discussion can be built with a wide range of clearances and still maintain highly efficient performance when pumping oil of a viscosity of 100,000 SSU. For example, a clearance anywhere from 0.01 to 0.05-inch will assure an efficiency, bearing friction neglected, of over 80 per cent. If a liquid with a viscosity of 10,000 SSU is to be pumped, clearances between 0.002 and 0.019-inch will assure an efficiency in excess of 90 per cent. Clearances between 0.05-inch and 0.095-inch will assure an efficiency in excess of 80 per cent with a viscosity of 1,000,000 SSU.

A number of other interesting points of comparison may be studied in *Figs.* 3 and 4. In all such studies it must be kept clearly in mind that the neglect of bearing and other friction gives higher indicated efficiencies than are actually obtainable. However, the performance curves are still comparable since all will be affected in a similar manner by the inclusion of all types of friction which are present in real pumps.

In connection with the possibility of building high-pressure, large-capacity, highly efficient positive displacement pumps, an interesting field which should be explored is the effect of overall size on efficiency

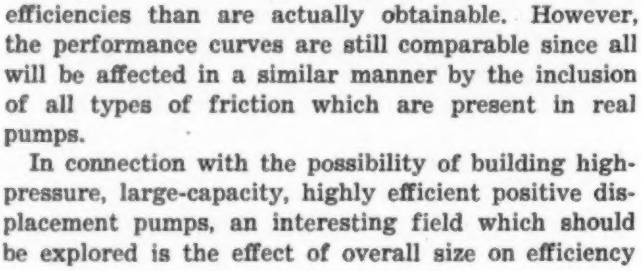

Fig. 4—Theoretical efficiency versus clearance for liquids of different viscosity ratings

and clearance requirements. It may be shown readily from a consideration of the equations for slip and viscous torque that geometrically similar pumps will perform with equal efficiencies under similar operating conditions. The operating conditions are fully specified by the dimensionless number $\mu_n/\Delta p$.

Consider the possibility of constructing a pump geometrically similar to the one previously under discussion but with all linear dimensions twice as great as those of the pump already designed. This new model would have a displacement of 1700 cu in. If operated at a speed of 1200 rpm, it would deliver approximately 8,000 gpm and would require about 6,000 hp to drive it. The clearances would be as follows:

Location	Clearance (in.)
Division members	0.068
Rotor ends	0.136
Vane ends	0.057
Vane tips	0.056

This pump offers almost fantastic possibilities. Its overall size would be small. The outside diameter need not exceed 30 inches and its axial length need not be more than 24 inches. It would be far smaller than any electric motor which could power it and would be smaller by a considerable amount than any gas turbine that might be large enough to turn its shaft at full load.

Hydraulic Balance Minimizes Loads

The problem of shaft and bearing loads is minimized by the fact that a double-acting vane pump has little or no load to be supported by the shaft and bearings. It is said to be "hydraulically balanced." It would doubtless be necessary to provide suitable bearings or rolling surfaces to permit easy sliding of the vanes. Alternatives are to use rotating vanes or to actuate mechanically the reciprocating vanes. Large loads on the vanes would doubtless necessitate the use of such mechanically driven vanes, since dependence on sliding action might well result in failure due to sticking of the vanes or the development of large friction forces at the sliding surfaces. In the conventional vane pump the frictional forces at the vane-to-rotor contact account for a large portion of the frictional loss in the pump.

In is pertinent at this point to determine the value of the pumps developed in this discussion. Although it is true that the design is not by any means complete, the essential facts concerning the significance of clearances in performance have been established.

The competent designer will notice immediately that the pump suggested here with large clearances exhibits certain mechanical weaknesses which would have to be corrected. For example, thickness of the vane is not sufficient to prevent bending under the load which would be imposed when operating at 1000 psi. Another point of weakness lies in the fact that the maximum diameter shaft, which could be used with the rotor and vane dimensions prescribed, would not be sufficient to transmit the torque unless a very

high-strength steel were used for the shaft. However, design details of this type can be corrected readily and no serious change in performance will be noted as a consequence. For example, making the vane several times as thick as indicated would result only in increasing the viscous drag and decreasing the slip, and neither to an appreciable extent in terms of total torque and slip. It should be understood that the purpose of this design analysis is simply to emphasize the importance and significance of clearances in the determination of the performance and not to provide a complete pump design.

The effect of the heating of the oil in the clearance spaces is a factor of major concern and no allowance for it has been made. It might be pointed out that performance of the pump with 0.005-inch clearance would be improved when operating with highly viscous liquids since the heating of the liquids would decrease the viscosity and give a performance characteristic corresponding to a lower viscosity.

It is to be hoped that pump designers will make use of the principles set forth here to improve the performance of pumps designed to handle extremely viscous liquids. Additional research directed toward the determination of the effect of temperature change on performance is in progress and should yield results of considerable benefit to the design of positive displacement pumps and motors.

REFERENCES

1. "Rotary Pump Theory"—W. E. Wilson, *Trans. ASME*, Vol. 68, 1946.
2. "Design Analysis of Rotary Pumps"—W. E. Wilson, *Product Engineering*, February, 1946.
3. "Hydraulic Pumps and Motors"—W. E. Wilson, MACHINE DESIGN, August, 1950.
4. "Performance Criteria for Positive Displacement Pumps and Fluid Motors"—W. E. Wilson, *Trans. ASME* Vol. 71, 1949.

Novel Belt Provides Precise Timing, Increased Power

IF the best features of several types of power transmissions could be combined into one simple drive, the resulting combination would be the answer to many a designer's problem. For instance, the low cost and flexibility of belt drives blended with the exact timing and higher power capacity of gearing might be the ideal drive solution for some applications. Such a drive, the Gilmer timing belt, has been employed for some time in special applications. Recently it has been made available for general machine application.

Similar in appearance to a flat belt but with the addition of regularly spaced rubber teeth along its inner surface, the belt permits exact timing between two shafts at belt speeds as high as 16,000 feet per minute. The teeth engage corresponding grooves in stamped or diecast pulleys. Strength of the belt is increased with a core of fine steel wire or, for light-duty applications, with a winding of cotton cord.

In addition to the positive drive and timing features, the belt is unusual in that power transmission capacity does not depend on high initial tension in the belt. This, of course, results in low shaft bearing pressures and long belt life. Low operating noise level is another characteristic of the drive. Speed ratios up to 30 to 1 are possible and belt flexibility allows use with pulley diameters as small as ½-inch at 10,000 rpm.

Applications of the belt have been made in which the timing, the capacity, and both have been the deciding factor in selection of the belt. Sewing machine drives require accurate timing with relatively high power, with repeated starting and stopping. Willcox & Gibbs Sewing Machine Co. has replaced the former flax belt with its metal clips by a ¾-inch wide wire cord timing belt transmitting ¼-hp at 5000 rpm motor speed and 2620 fpm belt speed. Operating speeds have been increased 50 per cent, with better synchronization between needle and bobbin shafts.

The portable hand planer illustrated below employs a timing belt to eliminate slip between the motor and cutting head. In this instance, the belt drive is more compact than gearing and requires no lubrication. A battery of wire-flattening machines with drives requiring 5, 7½, 10 and 15 hp uses Gilmer belts running at 16,000 fpm. Three-inch wide heavy wire belts are used on each drive, in spite of the varying power requirements, to simplify belt and pulley procurement.

As in any design problem, advantages of the timing belt must be weighed against its limitations. It is more sensitive to shaft misalignment than flat or vee belts, but less so than gears or chain. It is cheaper than the latter but somewhat more expensive than light-duty belt drives. As the power of a drive increases, the timing belt becomes relatively less costly as compared to standard belts. Although present timing belt drives are essentially custom-made at this time, the L. H. Gilmer Co. division of the U. S. Rubber Co. is considering a range of stock sizes in light and heavy-duty types.

Electronic Shear Control

. . . permits variations in cut lengths without mechanical adjustments

A NEW flying shear, left, incorporating photoelectric relays and timer circuits, serves the dual purpose of cropping the front ends of billets and then cutting them into hotbed lengths. The billet mill, at the delivery end of which the shear operates, is driven by an a-c motor which provides an essentially constant delivery speed of 295 fpm from the last stand. The shears are designed to cut all billets delivered from the mill up to and including two-inch squares.

Construction of the flying shear designed by Westinghouse Electric Corp. and United Engineering & Foundry Co. is somewhat unusual. The top knife, below, is attached to a crank crosshead and, similarly, the bottom knife and slide are attached to a second crank. To make a cut, each of the cranks makes one complete revolution. During this revolution or complete knife stroke, both knives are maintained at right angles to the bar by the slide and crosshead connecting the two shear heads on the cranks.

To minimize stresses on the knives, horizontal knife speed must match billet speed during the operation. Also, the shear must be stopped before making sufficient travel to cause a miscut. On these shears, the electrical drive and control allow for completion of the shearing cycle in 360 degrees of crank rotation, making it unnecessary to reverse the drive to return to the starting position. Because the cycle is completed in 360 degrees, no "reset" time is required.

Normally, the shear is completely automatic in operation. Either a photoelectric relay or a mechanical flag switch initiates the starting control. As soon as the shear starts, its control is arranged so that a suitable cam limit switch (which is geared to the drive) permits the shear to accelerate, make a cut, decelerate to a low creeping speed, and creep to the final stopping position—where it stops automatically. The control then resets, awaiting initiation of the next cycle.

The initiating device for making a front-end crop is located a few feet ahead of the shear. A similar device is located following the shear, to initiate starting of the shear to cut hotbed lengths. An electronic timer permits variation in hotbed lengths without physically moving the second initiating device.

In the event of a cobble at the mill, a manually operated pushbutton can be used to initiate the start of the shear. It will cause the shear to run continuously, as long as the button is depressed, and cut the cobble into relatively short lengths, which can be removed from the mill easily. As soon as the pushbutton is released, the shear completes its cycle and stops automatically.

Drive for the shear is a 50-hp, mill-type d-c motor, which in turn receives its power from a 50-kw adjustable-voltage generator. The motor fields are maintained constantly excited and all control sequences are obtained by variation of the generator field to provide adjustable-voltage control of the motors.

Selenium Rectifiers

... dependable, versatile, compact control units

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DURING the past five years, selenium rectifiers have been employed increasingly in control circuits requiring precise functioning. In addition to their primary use as rectifiers for supplying direct current, selenium cells are being utilized successfully as one-way valves for current blocking, nonlinear resistors and magnetic amplifier components. As rectifiers they are being applied over a voltage range from one volt to several thousand volts and from microamperes to thousands of amperes.

Fortunately the selenium rectifier came into being at a time when design engineers were turning toward electrical controls requiring economical and reliable sources of direct current. Even in its early stages of development, the selenium rectifier met these needs well enough to encourage further investigation. Compared with d-c sources then available, the selenium rectifier had advantages such as reliability, simple maintenance, high efficiency, accuracy of control, and compactness. The rapid strides made in development and production techniques in the past few years have done much to improve these characteristics further. Typical applications of these rectifiers are illustrated in *Fig.* 1 which shows two aircraft relays.

Fig. 1—Above—Two aircraft relays, manufactured by Advance Electric & Relay Co., and operated by full-wave rectified d-c from 400-cycle source, provide chatter-free operation and weight saving. The plug-in relay at right is designed for hermetic sealing

Before discussing industrial applications, a brief description of the selenium rectifier stack, the fundamental circuits, and the characteristicts of the rectifier should be helpful.

The basis of the selenium rectifier is a cell, made by coating a supporting metal back plate (usually aluminum) with a thin layer of metallic selenium and a front electrode layer. The rectifying junction or barrier layer is formed between the selenium and the front electrode. At this point rectification takes place. Current flows freely from the back plate to the front electrode, but is blocked in the reverse direction. Thus, the selenium rectifier cell is in itself a one-piece rectifier, requiring no high pressure contact for rectification.

Sizes and ratings of selenium rectifier cells vary

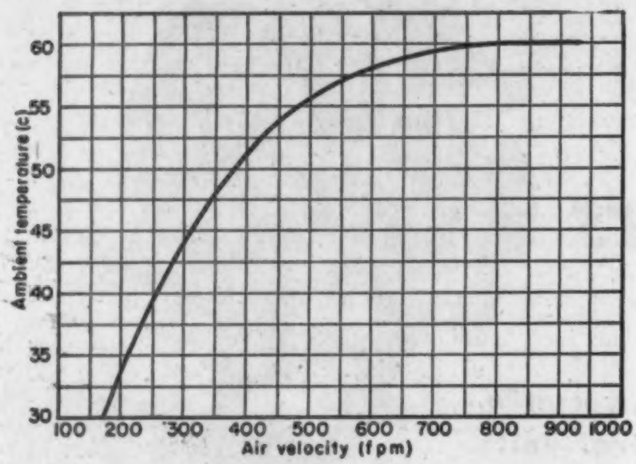
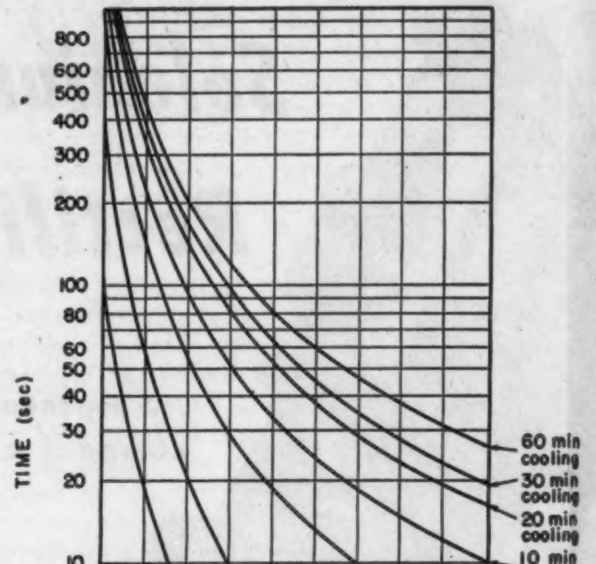


Fig. 2—Above—Minimum air velocities through rectifier at various temperatures when operating at 250 per cent normal current rating

Fig. 3—Right—Typical curves showing approximate time and overload factors to be applied for intermittent operation of selenium rectifiers at 45 C ambient



among different manufacturers, but standard Fansteel rectifier cells are made in circular sizes ranging from 9/32 to 4⅜ inches in diameter and in rectangular sizes up to 5 by 6 inches. They are rated at a maximum a-c input of 26 volts rms per cell, and current ratings, as shown in TABLE 1, are based upon continuous service at ambient temperatures up to 45. C.

A selenium rectifier stack is an assembly of cells connected in series, parallel, or series-parallel in various circuit combinations. The number and size of cells required in a given rectifier stack are governed by:

- Type of circuit
- Voltage (input and output)
- Load current
- Type of load
- Regulation
- Ambient temperature
- Duty cycle

RECTIFIER CIRCUITS: Six most commonly used rectifier circuits together with circuit formulas with constants sufficiently accurate for design purposes are shown in TABLE 2. Although the single-phase half-wave circuit is the simplest type of circuit, the ripple factor makes it unsuitable for many applications, and the utilization of the rectifier transformer is poor. In single-phase circuits, therefore, full-wave operation is generally used to attain greater utilization of the rectifier and transformer as well as a lower ripple factor.

Except in low-voltage applications, there is no difference in rectifier cost between a full-wave bridge and a full-wave center tap circuit. The single-phase full-wave circuit is generally suitable and commonly used for applications requiring up to one kilowatt of power. The ripple factor of this circuit is sufficiently low for the operation of d-c relays and other magnetic devices without additional filtering. On some applications (for example, when single-phase rectified d-c is used as a reference voltage in regulating circuits) a capacitor is connected in parallel with the load to reduce the voltage ripple.

On larger installations, requiring more than one kilowatt of power, or in applications where the percentage of ripple without a filter must be lower than that obtained from a single-phase circuit, three-phase rectifiers are used. The ratio of d-c voltage to maximum voltage for three-phase rectifiers is considerably higher than for single-phase units. Also, since each arm of a three-phase rectifier is conducting current for only 120 degrees, the total output current of the rectifier can be three times the average current rating of each arm. The utility factor and efficiency are higher than for a single-phase rectifier, so the three-phase rectifier is favored for most types of power applications.

REGULATION: Due to the nonlinear characteristics of the selenium rectifier, most of the output regulation occurs between zero and 20 per cent load. If conditions are such that the load is below 20 per cent and good regulation is essential, a bleeder resistor can be used to achieve fairly good regulation.

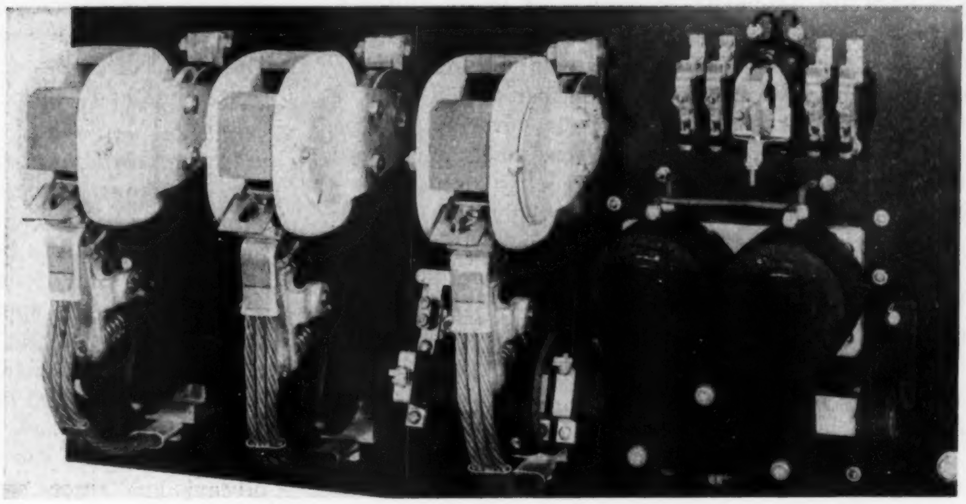


Fig. 4—Magnetic contactor is operated by a bipolar magnet powered by a selenium rectifier mounted on rear of panel

—Photo, courtesy Clark Controller Co.

In single-phase circuits, regulation is between 8 and 10 per cent from zero to the normal convection cooled rating. In three-phase circuits, the regulation is 4 to 5 per cent. When selenium rectifier stacks are built with wide spacing between the cells, or used with fan cooling, the regulation may be between 15 and 20 per cent.

When a capacity input-filter circuit is used, or when the rectifier works into a capacitor load, the capacitor is charged to the peak d-c voltage with each rise in voltage. While the d-c voltage drops to zero, the capacitor is discharged through the load and, if the load resistance is relatively high so that the rectified voltage sine wave drops faster than the capacitor discharges, the d-c voltage wave across the load will not drop to zero. This results in less voltage ripple, but the no-load voltage is equal to the peak d-c voltage and regulation between no load and full load may be relatively poor.

TEMPERATURE RANGES: There are many requirements for electrical equipment to operate over a wide range of temperatures and, in the case of rectifiers, stability is highly desirable. Unlike ordinary resistance devices, selenium rectifiers have a negative coefficient of resistance with respect to temperature. In other words, as the rectifier cell temperature rises, the forward resistance decreases, and the voltage drop across the rectifier is lower. Like any other current conductor, the current rating for a given active conducting area (usually expressed in square centimeters) is established by the operating temperature rise above ambient. Normal current ratings with ordinary convection cooling are therefore specified at a given ambient temperature, usually 45 C. When continuous service in higher temperatures is necessary, the rectifier must be used at a lower rating. Data for typical selenium rectifier ratings at elevated temperatures are shown in TABLE 3.

In applications exceeding normal current ratings, provision must be made to remove excessive heat. It is common practice, and especially economical on large units, to operate continuously at 250 per cent of normal current rating by means of forced cooling. Fan cooling is often employed to maintain safe cell temperatures in rectifiers operating in elevated ambient temperatures. *Fig.* 2 shows minimum air velocities at various air temperatures for one make of selenium rectifiers when operating at 250 per cent of normal rating.

There are many applications, too, where rectifiers

Table 1—Typical Ratings for Selenium Cells*

	Max. Continuous D-C Output†					Valve Operation				
	Single-Phase		Three-Phase							
Cell Diameter (in.)	Half-Wave (amp)	Bridge or Center Tap (amp)	Half-Wave (amp)	Bridge (amp)	Center Tap (amp)	Max Current (amp)	Blocking Voltage	Max A-C Voltage (rms volts per cell)	Max Cells per Stack (No.)	Cell Spacing on Centers (in.)
3/32	0.004	0.008	0.010	0.012	0.014	0.006	21	26	200	....
5/8	0.040	0.075	0.100	0.110	0.130	0.06	21	26	36	0.093
1	0.110	0.220	0.290	0.330	0.400	0.17	21	26	28	0.218
1 3/8	0.225	0.450	0.600	0.670	0.820	0.34	21	26	28	0.250
1 3/4	0.390	0.780	1.0	1.1	1.4	0.58	21	26	28	0.312
2 3/4	0.780	1.6	2.1	2.3	2.8	1.2	21	26	28	0.437
3 3/4	1.55	3.1	4.1	4.6	5.8	2.4	21	26	28	0.437
4 3/4	2.6	5.2	6.9	7.8	9.7	4.0	21	26	28	0.437

* Ratings are based on operation of Fansteel rectifiers at ambient temperatures of 45 C (113 F) or less.
† Continuous rectified output for selenium-cell stack with one cell per arm.

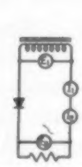
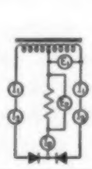
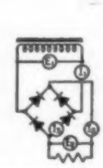
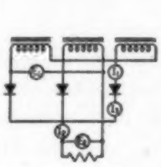
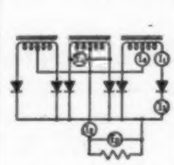
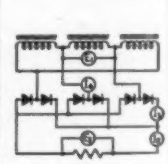
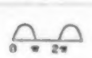
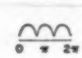
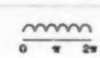
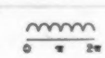
are used for intermittent service of short duration where it is not necessary for the design engineer to choose a rectifier having full current capacity. The active conductive area of the rectifier—or in simpler terms, the size of the rectifier—is governed by the time required for the cells to reach maximum safe temperature, the time allowed to cool to ambient temperature, and the voltage drop or regulation permissible in the circuit. The actual current rating in such instances is therefore based on the time cycle of the load. The maximum voltage ratings of cells, usually expressed in a-c volts rms, must not be exceeded.

Rating Depends on Duty Cycle

There are so many factors to be considered in this type of service that it is difficult to make any general set of rules. It is preferable to consider each application separately and, if necessary, make special tests to determine rectifier operating temperatures. The group of curves shown in *Fig.* 3 gives an approximate time and overload factor when the "off" period is known.

PROTECTIVE COATINGS: Selenium rectifier stacks normally have a painted protective coating to withstand average atmospheric conditions. Special protective coatings are available for especially adverse atmospheric conditions. At one time it was considered necessary to immerse rectifiers in oil to meet severe conditions but, with the choice of coated finishes now available, there is little if any advantage to immersion. An oil-immersed rectifier will require more mounting space, will generally be higher in cost, and will present the additional problem of possible oil leakage. Under some conditions where force cooling is required, the oil is circulated to transfer heat.

Table 2—Typical Rectifier Circuits and Characteristics*

	Single-Phase Half-Wave	Single-Phase Center-Tap	Single-Phase Bridge	Three-Phase Half-Wave	Three-Phase Center-Tap	Three-Phase Bridge
Diagram						
Output Voltage Wave shape						
Ripple Frequency	Fundamental	2 times fundamental	2 times fundamental	3 times fundamental	6 times fundamental	6 times fundamental
Ripple in Voltage	121%	47%	47%	18%	4%	4%
Approximate A-C:D-C Ratio	1.57	0.785	1.11	0.585	0.408	0.815
Min Cells Per Stack	1	2	4	3	6	6
Formula	$E_1=KE_2+ND$	$E_1=KE_2+ND$	$E_1=KE_2+2ND$	$E_1=KE_2+ND$	$E_1=2(KE_2+ND)$	$E_1=KE_2+2ND$
I_2	1.0	1.0	1.0	1.0	1.0	1.0
I_3	...	0.5	0.5	0.333	0.166	0.333
I_4	...	...	...	...	0.5	0.815
Resistance or Inductive Load:						
K	2.3	1.15	1.15	0.855	0.74	0.74
I_1	1.57	0.785	1.11	0.585	0.408	...
Output (rms amp)	1.57	1.11	1.11	1.02	1.0	1.0
Battery or Capacity Load:						
K	1.0	0.85	0.80	0.855	0.74	0.74
I_1	2.3	1.15	1.65	0.585	0.408	...
Output (rms amp)	2.3	1.55	1.65	1.02	1.0	1.0

E_1=Rectifier stack a-c input, volts
E_2=Output, d-c volts.
K =Circuit form factor

N =Number of cells in series in each arm of circuit
D =Voltage drop per cell at I_2 value

I_1=Rectifier stack a-c input, amp
I_2=Output, d-c, amp (average)
I_3=Amps, d-c, each arm of rectifier circuit
I_4=Phase amps, a-c

*The values will vary slightly with different circuit and load conditions, but are sufficiently accurate for design purposes.

—Photo, courtesy Cutler-Hammer Inc.

Fig. 5—Left—Selenium rectifier in high tension control cubicle supplies power for magnet coil of oil-immersed high tension contactor controlling squirrel-cage motor drive

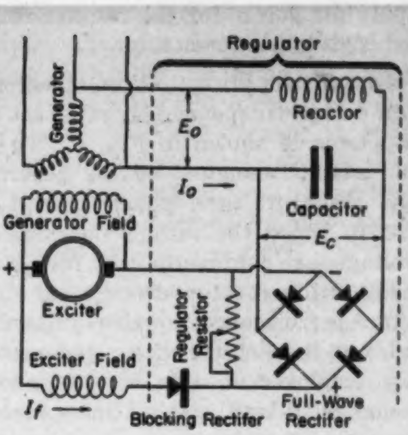


Fig. 6—Right—Circuit diagram shows use of selenium rectifiers to regulate voltage of high-frequency generator

Since many atmospheric conditions may be harmful to a rectifier improperly protected, the design engineer should present all the facts concerning atmospheric conditions to the manufacturer for recommendations as to the type of protection required.

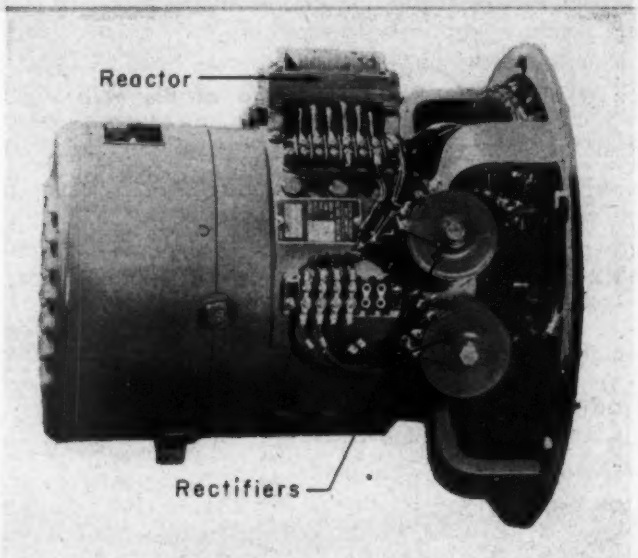
Provides Better Performance

Relays and Contactors: In *Fig.* 1 are two types of d-c relays, one of them hermetically sealed, operated by full-wave rectified d-c from a single-phase 400-cycle a-c source. The small rectifiers are mounted as component parts of the relays. The reasons for using a d-c relay with a rectifier rather than an a-c relay in this application are to obtain:

1. Positive, chatter-free operation
2. Better control of temperature rise in the relay coil
3. Smaller, lighter weight relay for aircraft use.

It is difficult to build satisfactory a-c magnets for the operation of large contactors and it is equally difficult to avoid the inherent magnetic hum in large a-c magnet operators. A magnetic contactor, operated by a single-phase full-wave bridge-connected rectifier from 110 volts a-c, is shown in *Fig.* 4. The magnetic contactor is rated at 600 amperes per pole. The contactor is operated by a bipolar d-c magnet with an operating coil on each pole. The coils, which are series connected, are supplied with d-c power from the selenium rectifier with a maximum current rating of 1.2 amperes. Full-line voltage is applied to the rectifier to close the contactor, then a series resistance is inserted to reduce the operating coil voltage to a suitable holding value when the contactor has been closed.

The rectifier usually can be designed to operate from the a-c line without a transformer. Either a small, compact rectifier for each contactor, or a large rectifier power unit for several contactors provides an economical source of d-c control power. In addition, d-c operation provides inherent noiseless operation for the life of the equipment. *Fig.* 5 shows a selenium rectifier stack mounted in a control cubicle

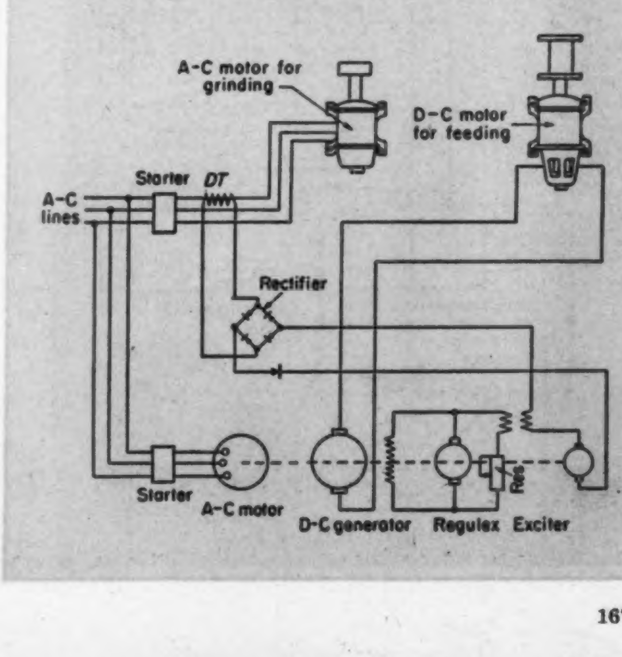


—Photo, courtesy Electric Machinery Mfg. Co.

Fig. 7—Above—Rectifiers and other control components mounted on frame of exciter provide voltage regulation to high-frequency generator

Fig. 8—Below—Selenium rectifier is essential part of circuit controlling speed of feed motor on grinder

—Diagram, courtesy Allis-Chalmers Mfg. Co.



to supply d-c power for the magnet coil of an oil-immersed high-tension contactor.

Voltage Regulation: A voltage regulator circuit used on a high-frequency generator set for high-speed electric tools is shown in *Fig.* 6. The rectifiers are shown "saddle" mounted on the generator exciter in *Fig.* 7. Miniature three-phase squirrel-cage induction motors, in which the torque varies as the square of the voltage, are commonly used for grinding, buffing, polishing, drilling, screw driving and nut running. For good tool performance, a well-regulated voltage is essential, but the voltage of a-c generators of the type usually employed in this service tends to fluctuate with change in load, so that some form of automatic voltage regulation is necessary.

The voltage regulating circuit shown in *Fig.* 6 is a resonant circuit at 60 cycles. The generator a-c output is fed into the regulating circuit, and the resonant voltage is taken across the capacitor, rectified through a single-phase full-wave bridge rectifier, and applied to the generator field in opposition to the regular field. Thus, the net voltage operating on the exciter field is the difference between the opposing rectifier and exciter voltages. When the output voltage of the generator increases, the voltage across the capacitor increases in much greater proportion, and this is applied, after rectification, to the field of the generator. This rectified voltage opposes the voltage applied to the exciter field winding by the exciter, reducing the magnetic field flux of the exciter and the exciter voltage and, in turn, reducing the voltage across the field winding of the generator and the generator voltage. When the output voltage of the generator decreases, the action is reversed. The additional half-wave rectifier shown in the circuit acts as a valve to permit flow of d-c in one direction only to prevent reversal of polarity of the excited field current.

It would be impossible to cover in detail all the different circuits in which selenium rectifiers supply a d-c reference or bias voltage, or where they are used as valves for voltage, current or speed control. A feed control on a chain grinding machine, allowing chain to be fed into the grinder at a maximum rate of speed compatible with the full load of the grinder motor, is illustrated in *Fig.* 8 and a current-limit control circuit is shown in *Fig.* 9.

Current Blocking: Use of a half-wave rectifier as a valve or blocking rectifier in a d-c control circuit is shown in *Fig.* 10. In the left hand drawing, the polarity is such that current flows through the rectifier and energizes the relay. In the drawing at the right, the d-c polarity is reversed and the rectifier

Table 3—Rectifier Rating at Elevated Temperatures

Ambient Temperature (C)	(F)	Current (% Normal)	Voltage (% Normal)
Up to 45	113	100	100
50	122	95	100
50	122	100	95
55	131	90	100
55	131	100	88
60	140	82	100
60	140	100	75
65	149	75	100
65	149	100	62
70	158	70	90
70	158	90	58
75	167	65	80
75	167	80	55
80	176	55	65
80	176	65	45
85	185	43	50
85	185	50	35
90	194	32	45
90	194	45	25

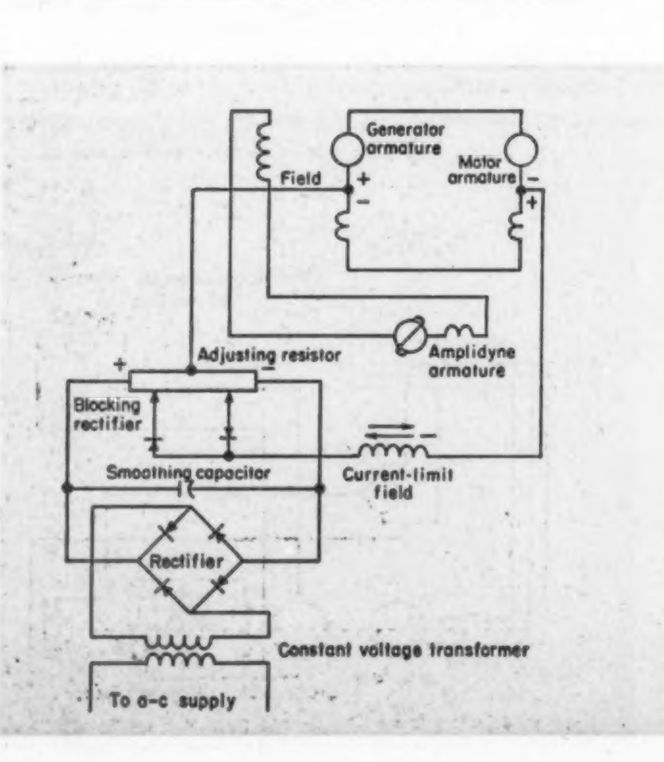
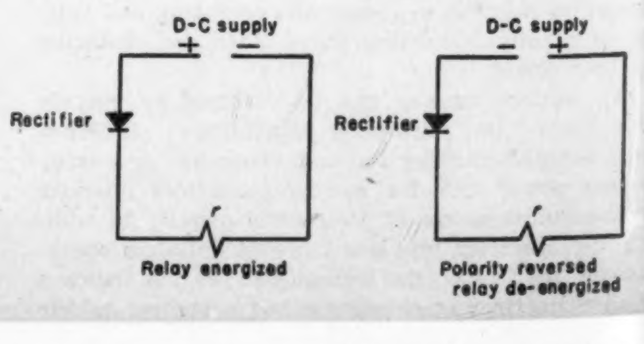


Fig. 9—Left—Current-limit control for motor-generator circuit, using full wave selenium rectifier and two half-wave blocking rectifiers

Fig. 10—Below—Use of blocking rectifier in d-c relay circuit. At left, current flows through rectifier, energizing relay *R*. At right, when d-c polarity is reversed, rectifier blocks current, and relay is de-energized



blocks the flow of current through the relay coil, thus de-energizing the relay. Blocking rectifiers, therefore, are used in control circuits to prevent the flow of current in one direction if the polarity should be reversed. This application is commonly used in relaying circuits. In blocking applications, special ratings are applied and are based on the percentage of time the rectifier is carrying current compared with the time it is blocking.

Braking Circuit Is Simple

Dynamic Braking: Squirrel-cage motors can be retarded by disconnecting the stator from the line and exciting the stator with direct current. A simple control circuit for dynamic braking a squirrel-cage motor to a stop is shown in *Fig.* 11. When the motor is running, contactor *M* is normally closed and *DB* is open. When the "stop" button is pushed, dynamic braking is applied, dropping out contactor *M* and closing *DB*, connecting a selenium rectifier to the line and feeding d-c to the motor stator. Since the d-c resistance of the stator winding is relatively low, a series resistor is used as a current limiter.

Most applications of this type are intermittent duty for the rectifier and can be designed for intermittent service, as described previously. From a voltage standpoint, the rectifier may be designed to operate direct from the line voltage, except on high voltage motors where a transformer may be employed to permit the use of a more economical low-voltage rectifier.

The rear of an adjustable voltage control panel for a supercalender drive is shown in *Fig.* 12. One of the rectifiers on this panel controls a voltage limiting field of a rotating regulator. The other acts as a blocking rectifier in the field circuit of the reel motor to prevent the field from becoming weakened to an extent which would cause overspeeds of the reel motor.

An elementary diagram and a photograph of a magnetic brake controller, showing a single-phase bridge rectifier and a relay which controls a d-c brake solenoid, are shown in *Fig.* 13. *Fig.* 14 shows the same application including a d-c operated timing relay which inserts a protecting resistor in the brake coil circuit. This design permits the use of a smaller and more economical selenium rectifier.

Nonlinear Resistance: Typical current flow characteristics of a selenium rectifier are shown in *Fig.* 15. The current is zero at zero voltage and resistance is at a maximum. Resistance falls rapidly as voltage is first applied in the forward direction and then tends to approach a constant value. Resistance in the reverse direction also decreases with increasing voltage, but the rate of decrease is much less rapid. These characteristics lead to the use of selenium rectifiers as nonlinear resistance devices, regulators, clippers and arc suppressors.

There are many occasions when it is desirable to

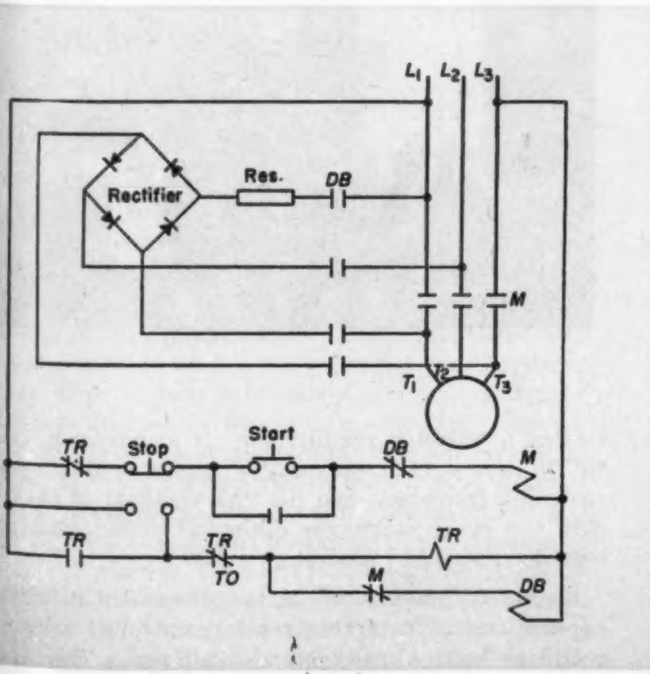


Fig. 11—Left—Diagram for d-c dynamic braking control circuit for squirrel-cage induction motor, using single-phase full-wave rectifier without transformer

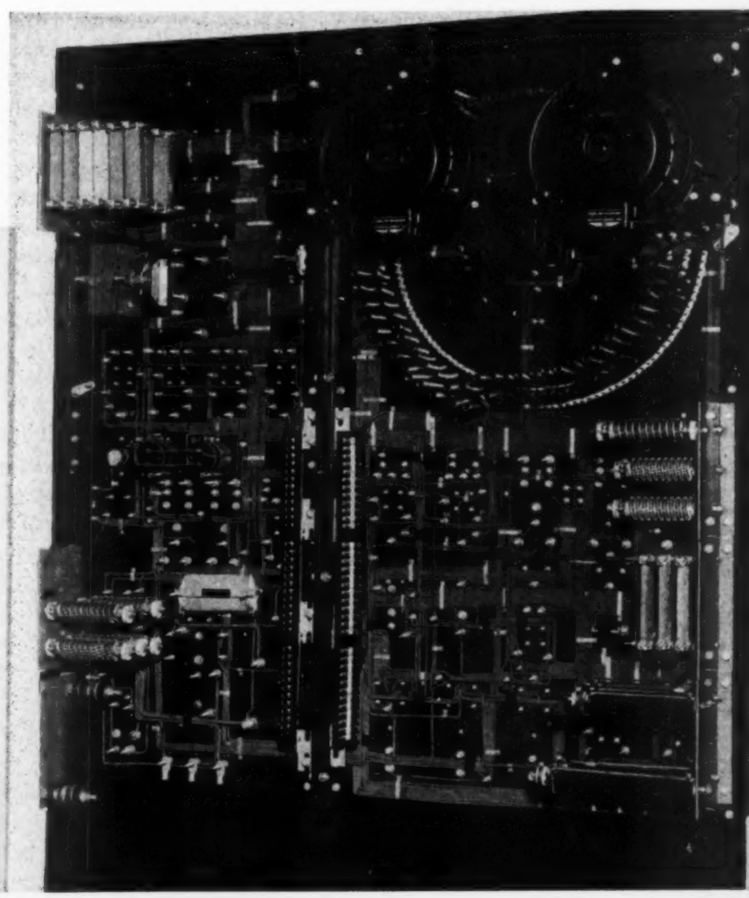
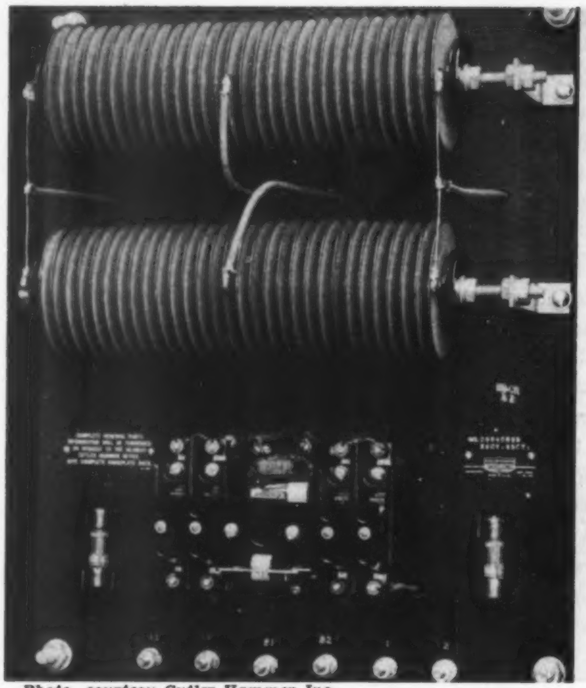


Fig. 12—Below—Adjustable control panel (rear) for supercalendar motor. One selenium rectifier controls voltage limiting field of rotating regulator. The other, a blocking rectifier, prevents dangerous overspeeds

—*Photo, courtesy Cutler-Hammer Inc.*



—Photo, courtesy Cutler-Hammer Inc.

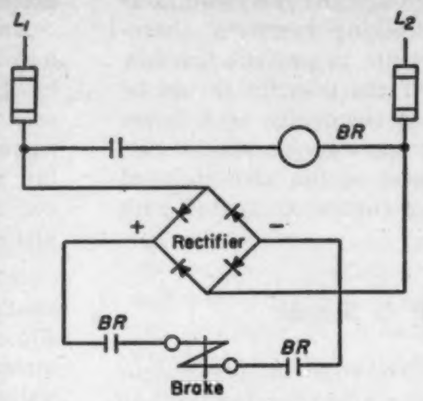


Fig. 13 — Left — Magnet brake controller, photo graph and diagram, show ing single-phase bridge re tifier which supplies pow to brake solenoid

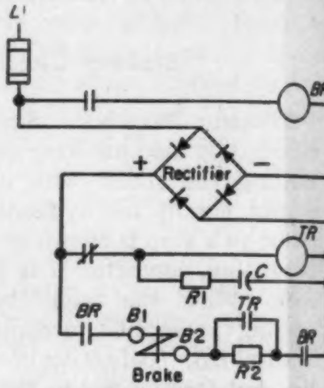
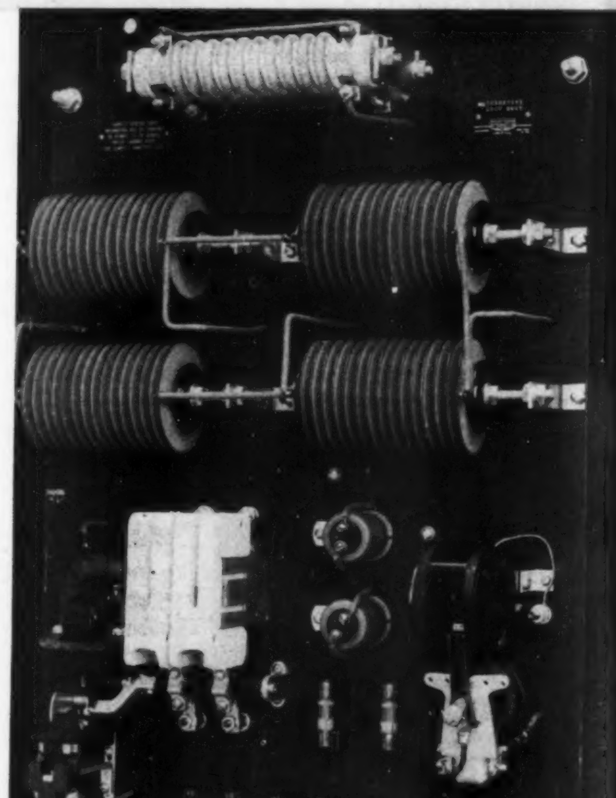


Fig. 14—Right and Below —Magnetic brake application, similar to Fig. 13, employs resistor inserted into brake coil circuit by timing relay, permitting smaller rectifier



—Photo, courtesy Cutler-Hammer Inc.

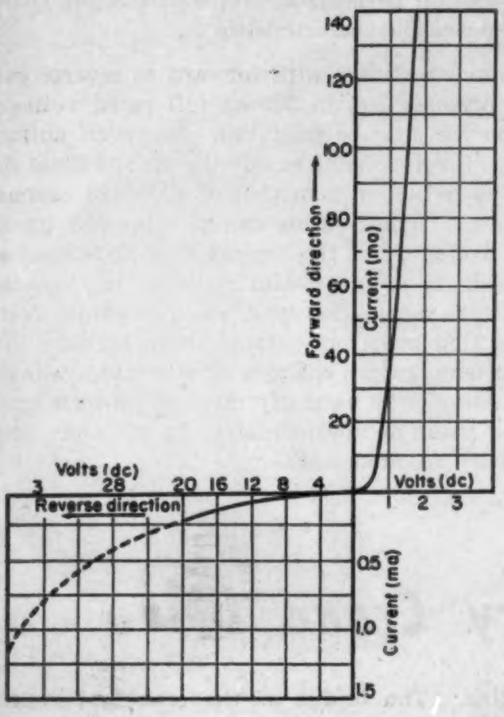
use a nonlinear resistance device for regulating voltage. A schematic diagram for connecting a half-wave rectifier to provide voltage regulation in a d-c circuit is shown in *Fig.* 16. Referring again to *Fig.* 15, it will be seen that considerabe regulation can be obtained by operating the rectifier in the steep portion of the curve. Best performance is usually attained by operating within the range of 0.05 to 0.6-volt d-c per series junction. Because of the voltage limits within which this type of regulator must be operated, it is particularly useful in low-voltage circuits. Since it requires a rectifier with a relatively large number of series junctions, it is not economical for high-voltage applications.

Selenium rectifiers for protecting electrical contacts against the destructive effects of arcing are receiving increasing attention. Arcing of contacts is especially severe in inductive d-c circuits, and the degree to which arcing can be suppressed or quenched is probably the most influential factor governing the life of the contacts. A rectifier connected in parallel with the inductive load, as indicated in *Fig.* 17, is effective in reducing or entirely eliminating the arc in many applications. The desirable features of this type of quench circuit are:

1. Effective prevention of arcing in closing and opening of contacts
2. Satisfactory rate of current fall, compared with unquenched conditions
3. Dependability over extended periods of operation without attention
4. No arcing of contacts from condenser charge or discharge
5. Small mounting space for the rectifier
6. Relatively low cost.

A number of factors must be considered when designing a selenium rectifier for arc suppression, such as the value of steady-state voltage and current, switching frequency, and the time constant of the circuit. In many cases, the optimum rectifier size has been determined by laboratory tests.

Magnetic Amplifier: A few diversified industrial applications of saturable reactor-controlled selenium rectifiers have already been mentioned. This com-



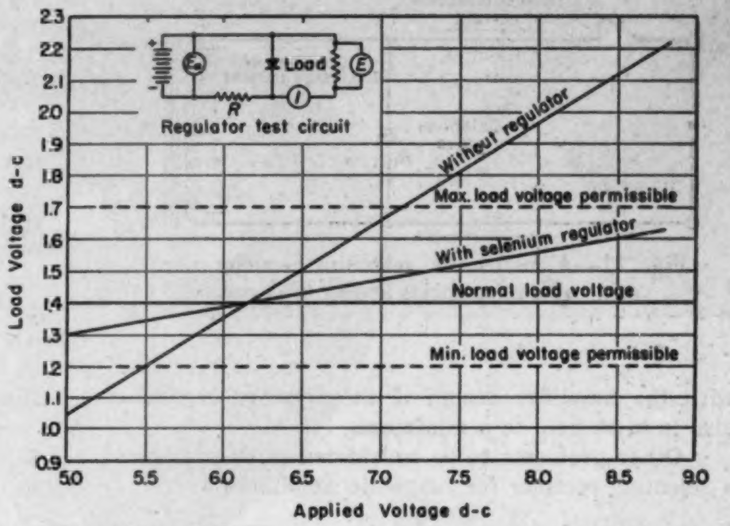


Fig. 15—Left—Forward and reverse current flow characteristics of a typical selenium rectifier cell having an effective rectifying area of one square centimeter

Fig. 16—Above—Selenium rectifier used for regulation of d-c load of 0.06-ampere from applied potential of 5 to 10 volts and a load potential of 1.4 volts. Test circuit is shown in inset

bination of the selenium rectifier and a saturable-core reactor, now commonly called the magnetic amplifier, is becoming increasingly popular for use in regulating and control circuits. Increased interest in magnetic amplifiers in recent years may be attributed to the need for improved and maintenance-free static components for control circuits, and further understanding of the self-saturating circuit. Recent developments in magnetic core materials and selenium rectifiers have accelerated this development, and indeed are largely responsible for the successful performance now being obtained.

Circuits for Magnetic Amplifiers

Most standard rectifier circuits can be converted to self-saturating magnetic amplifier circuits by the addition of a saturable reactor. Either single or polyphase circuits may be used, and the normal expressions for voltage and current relationships in a rectifier circuit are reasonably accurate.

Many different circuits for magnetic amplifiers may be used for a d-c amplifier by feeding the d-c signal voltage directly into the control winding, and then rectifying the a-c output of the amplifier. An a-c amplifier may be obtained by rectifying the a-c signal voltage, applying this voltage to the d-c control winding and taking the a-c output from the amplifier. Because of the ripple in the rectified voltage applied to the control winding, the performance of the magnetic amplifier may be slightly different from that obtained when pure d-c is used.

Feedback in magnetic amplifier circuits is also commonly practiced. The output current is rectified in a full-wave bridge rectifier and introduced in a special feedback winding on the reactor in a manner to aid the control circuit. Either positive or negative feedback may be used, depending upon conditions. The use of feedback results in increased sensitivity, and extends the range further than that of a simple reactor.

Performance of a properly designed magnetic amplifier system depends upon the properties of the core material and the series rectifier. Low forward resistance and high reverse resistance (low leakage) in the rectifier, consistent with the properties of the magnetic core material, are desirable for obtaining maximum sensitivity. Because of the relatively large number of turns in the a-c windings, even a small leakage current may require a large increase in the control equipment. With the high sensitivity possible in new type cores, the series rectifier must have correspondingly low leakage if optimum performance is to be obtained.

At present, standard commercial metallic rectifiers are not suitable for use with high-permeability core materials. Germanium crystal diodes are suitable for extremely low current applications but have a relatively high forward resistance. For larger applications, specially made selenium rectifiers are the only type of metallic rectifier in which reverse resistance approaching desirable limits can be obtained. Even with these special rectifiers, it is necessary to permit an increase in forward resistance in order to obtain optimum leakage characteristics. In specifying such rectifiers, it therefore becomes necessary to specify minimum forward to reverse current ratios consistent

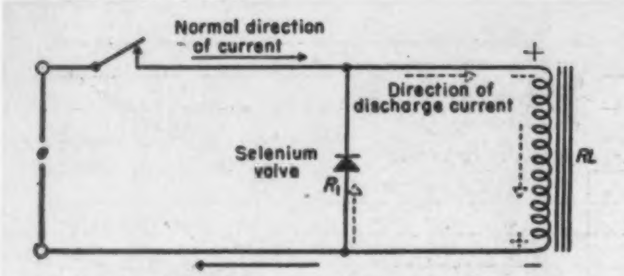


Fig. 17—A half-wave selenium rectifier used as a valve to suppress arcing of contacts

with the amplifier design if the forward resistance also is to be held to a minimum.

Other problems to be considered in the design of a selenium rectifier for magnetic amplifiers are:

1. Stability over a wide temperature range
2. Rapid recovery of original characteristics after equipment idleness
3. Reproducible characteristics.

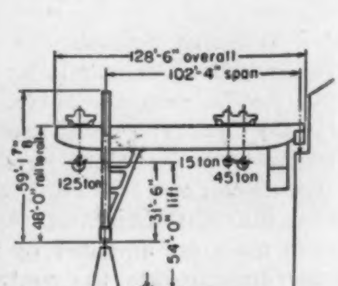
Selenium rectifiers with forward to reverse current ratios between 200 to 300 at full rated voltage per junction can now be produced. However, uniformity is still difficult to achieve and the cost of these special rectifiers is higher than that of standard commercial rectifiers. Higher ratios can be obtained by derating in voltage, but this means that additional series junctions are required with resulting increase in forward resistance. However, since selenium rectifiers of this kind have more stable characteristics in relation to temperature changes at a derated voltage, the operation of cells normally rated at 26 volts rms at a derated value of approximately 15 volts per junction is usually recommended.

Nonsynchronized Gantry Crane Drive

UNUSUAL not only because it is the biggest of its type but also because of its shaftless bridge drive, the 125-ton single-leg gantry crane shown in the accompanying illustration was built by The Cleveland Crane & Engineering Co. to serve Bessemer converters at The National Tube Co. in Lorain, Ohio.

Heretofore, bridge drive motors for all long-span semigantries have been mounted on the bridge girders and arranged to drive the track wheels through line shafting, flexible couplings, bevel gears and universal joints. On this new crane, however, individual motor bridge drives are utilized, welded box-section portal type legs providing maximum torsional rigidity and assuring proper alignment of all parts for successful operation. The bridge of the crane rides on eight equalizing type two-wheel trucks, four of which are located at the gantry leg and four at the wall end of the crane girders. For each pair of trucks there is a direct-connected individual motor drive, making four bridge drive motors in all. No provision is made to electrically synchronize the motors, resulting in a simpler drive.

The hermetically sealed welded box sections used for both girders and leg provide maximum resistance to corrosion, always a problem in the Bessemer area. Dimensions of the crane are shown in the inset sketch. One trolley is equipped with a 25-ton hoist, the other with 15 and 45-ton hoists.



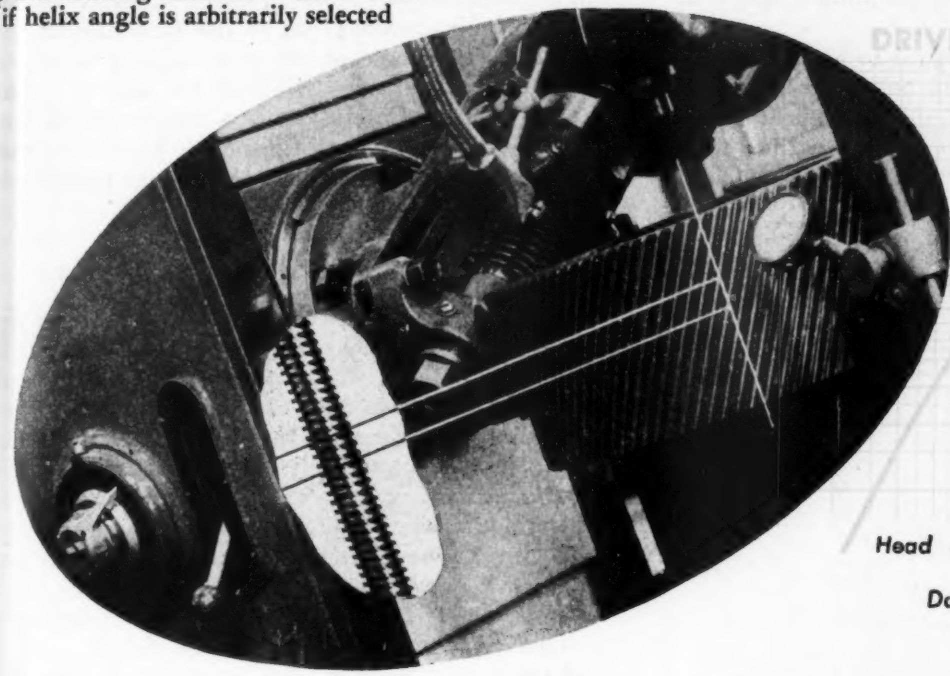


Fig. 1—If the axial pitch of a helical gear is made a simple multiple of the axial pitch of the feed screw, setting up the hobbing machine is easier than if helix angle is arbitrarily selected

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What Helix Angle

. . . for hobbed helical gears?

HELICAL gears that are to be cut by the hobbing process and are to mesh on parallel shafts may be designed to work at any one of a wide range of helix angles. What are the restrictions on helix angle? Are any angles within the permissible range to be preferred to any others? Answers to these questions are suggested in part by the nature of the hobbing process, *Fig.* 1.

First let it be made clear what is meant by *helix angle*. It defines the direction, at any particular point, of the line of intersection of a tooth-flank and an imaginary cylinder coaxial with the gear. The line of intersection is called the *trace* of the tooth flank on the cylinder concerned. The angle between the tangent to the trace at any point and the straight line that contains that point and lies parallel to the axis of the gear is the helix angle at that particular cylinder. At any other cylinder a different helix angle will be found. To speak of helix angle is therefore meaningless unless some cylinder is specified, implied or understood. In this article, *helix angle* means the helix angle at the pitch cylinder of generation of the gear teeth by the hob—which is not necessarily the same as the pitch cylinder of engagement with the mating gear.

If the normal pitch, p_n, of the teeth is known, the helical characteristic of a gear may be defined by the axial pitch, p_x. This is the distance between corresponding flanks of adjacent teeth measured along a line parallel to the axis of the gear. This is a dimension of greater fundamental significance than any helix angle, because the axial pitch is the same for all possible distances of the line of measurement from the axis of the gear. Relationships of the various pitches and helix angle are shown in terms of the basic rack in *Fig.* 2.

Furthermore, the axial pitch is of practical importance for it is the dimension used in the calculations for setting up the gear-hobbing machine to produce a helical gear of specified form. Often the formula for change-gear ratio is based on the *lead* of the gear, but the lead is simply the axial pitch multiplied by the number of teeth.

Tedious arithmetical work is involved in discovering change-gear combinations that have irrational ratios such as 6.3819. Actually there is never any need for such a ratio in the spiralling change gears of a hobbing machine if gears to be cut are intelligently designed.

In helical gears of given materials, diameters and widths, variation in helix angle has no effect on the load capacity as limited by surface pressure on

the teeth. Limitation of load capacity by bending stress can always be avoided by adopting a sufficiently coarse pitch.

A helical gear may be designed to have any helix angle between 90 degrees right hand and 90 degrees left hand, but not every part of this range is advisable. There is some evidence that for a given normal pitch, increasing the helix angle to 45 degrees, at least,

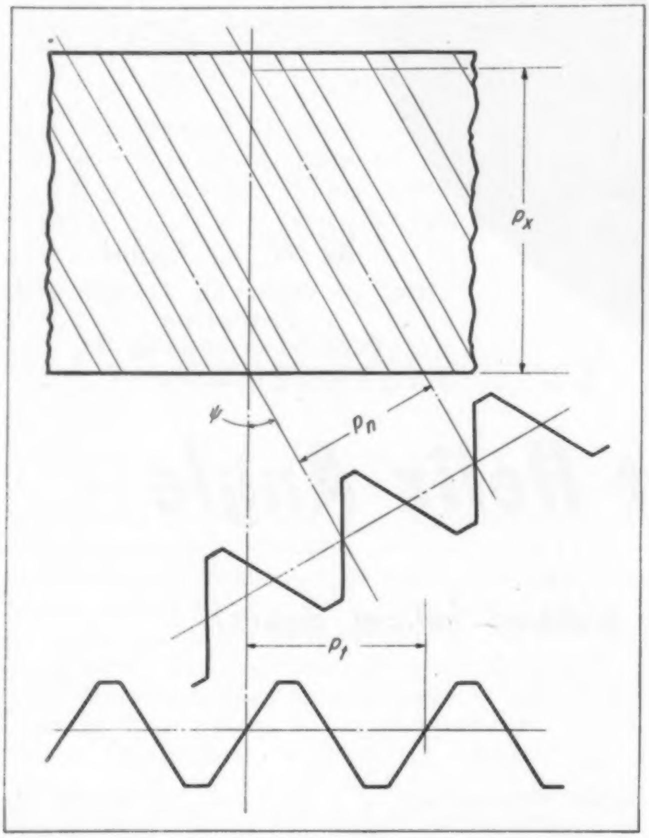


Fig. 2—Basic dimensions shown on this helical rack are helix angle ψ, axial pitch p_x, normal pitch p_n and transverse pitch, p_t

leads to quieter running. If the diameter of the gear and its number of teeth are fixed, increase in helix angle reduces the normal pitch of the teeth and increases the normal load on them, so increasing the bending stress. On this account a helix angle near 45 degrees demands a coarser transverse pitch, p_t, than will suffice for, say, 30 degrees. The corresponding gain in adopting a helix angle much less than 30 degrees is not great. For double-helical (herringbone) gears, which exert no end thrust on their shafts, a helix angle between 25 and 35 degrees is therefore as good as any other.

For single-helical gears, desire to keep down the end thrust suggests a smaller helix angle. Any angle down to 15 degrees may be used. In no case, however, should the axial pitch be less than the axial width (face width) of one helix. This would result in incomplete overlap of tooth action at one end of the face width in relation to that at the other end and the full advantage of the helical tooth form would not be obtained.

It is found advantageous to use three-thread hobs to cut a helix angle of about 23 degrees in double-helical gears that are required to have a central gap of minimum width. Such a hob, made of minimum diameter solid with its spindle, has a lead angle of about 23 degrees and therefore lies at right angles to the axis of the work when cutting.

Preferred Helix Angles for Hobbing

It is seen, therefore, that for three different purposes approximate helix angles of 30, 23 and 15 degrees are suitable. The following tabulations give helix angles corresponding to axial pitches that can be cut on hobbing machines set up with spiraling gears of exact ratio, with no calculation needed in selecting change gears. The tables are based on the assumption that the pitch of the feed screw of the hobbing machine is a simple fraction of an inch, such as ½ or ¾, and that the normal pitch of the hob is either a simple fraction or multiple of an inch or is a standard diametral pitch, such as 4 DP.

If the normal pitch, p_n, is a standard circular pitch, that is, a simple fraction or multiple of an inch, axial pitches, p_x, that should be specified and their corresponding helix angles are as follows:

Axial Pitch	Helix Angle
$4\ p_n$	14° 30′
$2.5\ p_n$	23° 30′
$2\ p_n$	30°

If normal pitch is a standard diametral pitch, P_n, the following axial pitches and corresponding helix angles should be specified:

Axial Pitch	Helix Angle
$12/P_n$	15° 12′
$8/P_n$	23° 10′
$6/P_n$	31° 34′

It may be added that there is no object in specifying the helix angle any more precisely than is done in these tables. The helix angle used in the angular setting of the hob spindle should be halfway between the helix angles at tip and root of the tooth. This is not the same as the helix angle of generation, although the difference usually is negligible. The tangent of the helix angle at any cylinder coaxial with the gear is equal to the circumference of that cylinder divided by the lead of the teeth.

Correction

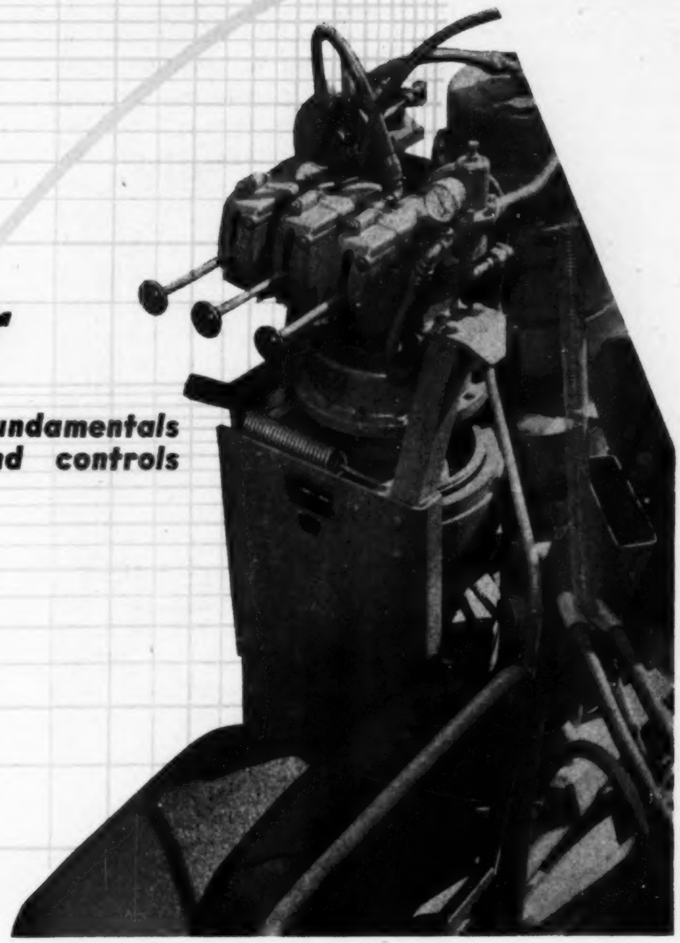
In the article "Determining Practical Tolerances" by Wayne A. Ring which appeared in the March issue, several typographical errors escaped notice. Following are corrected versions of two sentences from page 123 of the article:

"The chances of both happening at once are $(c-x_1)\Delta x_1/(c-a)(d-b)$."

"By the use of $T_A = 6\sigma_A$, $T_B = 6\sigma_B$ and $M_O = ½ (T_A + T_B) - T_O$, this can be transformed to . . ."

Pneumatic Power

Design and application fundamentals for machine drives and controls



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Part 1—General Considerations

COMPRESSED air is one of the most versatile mediums for powering and controlling machine elements, *Fig.* 1. Its field of application is so varied that all machine designers should have a working knowledge of pneumatics for coping adequately with power and control problems. Toward this end this series of three articles will present some of the fundamental factors for designing various types of systems.

AIR POWER: Pneumatic power can be applied through air jets, rotary motors or linear actuators. The most valuable characteristics of compressed air for these uses are that it is clean, abundant in supply, makes no mess if spilled or wasted, and will not adulterate the materials with which it is mixed.

The rotary air motor has particular characteristics which make it suitable for many applications. It is small and lightweight, quickly and easily adjusted to a wide range of speeds and unhurt by stalling. These characteristics make it valuable for hand grinders, drills, and other portable tools. Its ruggedness, lack of explosion hazard, and serviceability in damp and corrosive surroundings make it particularly suitable in mines or other places where electrical equipment is difficult to maintain. However, the air motor cannot compete with the electric motor for ordinary uses where power cost is more important than the foregoing factors. This is because the electric motor is inherently more efficient than the air motor.

Linear actuators, which include a wide variety of compressed air cylinders and positioners, are the most important of the pneumatic power-producing devices. They can be used to operate valves, gates, doors, switches, clamps, lifts, punches, brakes, clutches, throttles, and many other devices. It is this group of actuators and their operating and controlling valves that make up the vast majority of pneumatic control systems. The application of air power to brakes, clutches, and throttles will serve to illustrate several of the basic types of linear actua-

Fig. 1—Above—Operator's station on Wooldrige Mfg. Co. earth mover showing pneumatic controls

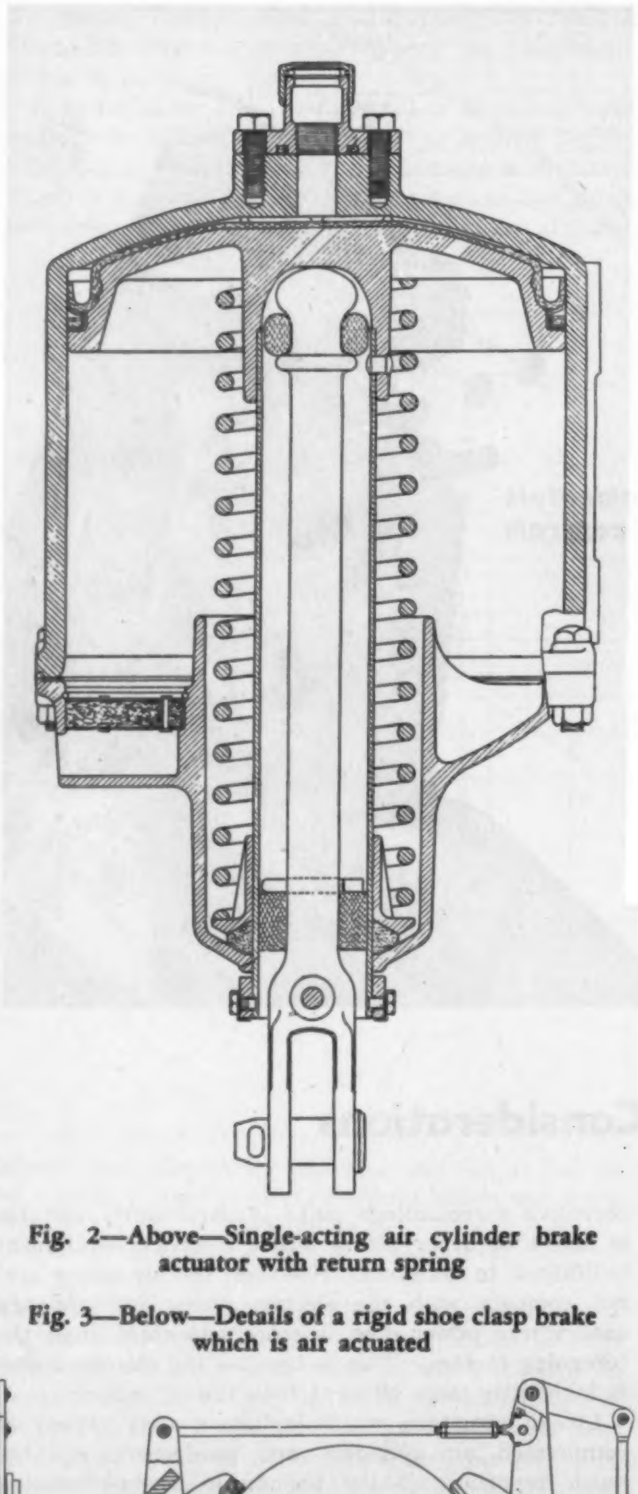
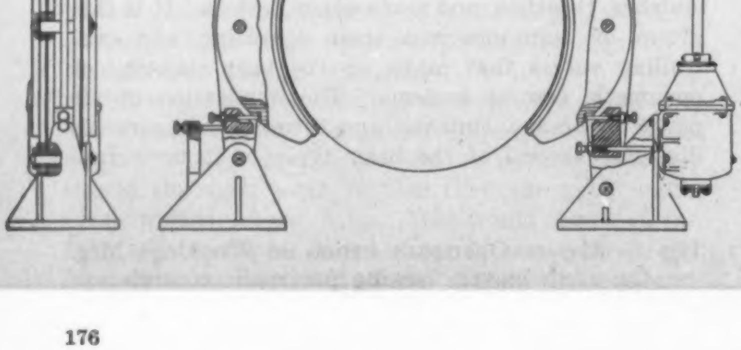


Fig. 2—Above—Single-acting air cylinder brake actuator with return spring

Fig. 3—Below—Details of a rigid shoe clasp brake which is air actuated



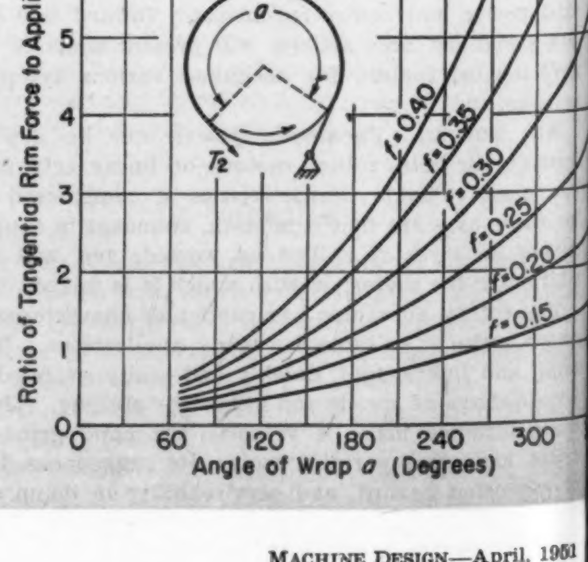
tors and their characteristics.

Cylindrical Actuators: A single-acting air cylinder with a return spring as shown in *Fig.* 2, is a typical brake actuator. The brake is applied by supplying air pressure to the cylinder, the braking force being varied by control of the pressure in the cylinder. This results in flexible control of the resulting brake torque only if the brake itself is designed to have an output torque proportional to operating force. Typical of this type of design is the clasp brake illustrated in *Fig.* 3.

A self-energizing brake such as a flexible band around a brake drum will have considerably less flexibility when power controlled. *Fig.* 4 shows this type of brake and illustrates the factors affecting self-energizing or "wrap-up" effect. For good power control of a band brake the coefficient of friction should be low enough and the angle of wrap small enough that the factor P/T_2 is less than 2.5. Other types of pneumatically controlled brakes are the spring-applied, air-released brake; and the mechanically applied brake with a pneumatic booster.

Clutch operating cylinders are similar to brake cylinders except that they do not always require flexibility of control. Sometimes they involve an on or off control with possibly an orifice in the air line to control the rate of clutch engagement. Where forward and reverse clutches are controlled by a

Fig. 4—Typical flexible band brake and chart giving characteristics



single operating lever, a three-position cylinder device such as the Tridyne Positioner, shown in *Fig.* 5, is used. The double-acting cylinder is also used on clutches where considerable force is required to disengage the clutch. On many marine clutches an "over center" linkage is used to keep the clutch engaged. The operating cylinders on these clutches are automatically exhausted of pressure after clutch engagement so as not to cause rapid wear of the thrust collar in the clutch.

Throttle Controls Govern Accurately

Remote control of throttles and governors on gasoline or diesel engines usually requires a positioning actuator capable of producing varying throttle settings in response to a varying pilot pressure from the control valve. The simplest type of throttle actuator, as shown in *Fig.* 6, consists of a diaphragm with a heavy calibrated spring on one side and variable pilot pressure on the other. A change in pilot pressure causes a corresponding spring deflection with the result that diaphragm movement is directly proportional to pilot pressure. If the spring force is large compared to operating force of the throttle, the diaphragm movement can be used to accurately position the throttle. More powerful types of pneumatic positioners are required if the operating force of the throttle is large.

Remote and Power Controls: Illustrating the use of these actuators and their associated valves, the railway air brake is one of the best known pneumatic systems. A 150-car freight train has a 10-inch diameter air cylinder on each car to apply the brakes. The cars are spread over more than a mile and a half distance and must respond to movement of the brake valve on the locomotive through a single pipe the length of the train. By controlling the pressure in this pipe, all 150 brakes can be applied to any degree or released. Through the same pipe air is charged into the local storage reservoir on each car. The transmission of braking impulse through this pipe can exceed a rate of 900 feet per second. These pneumatic brakes can be interlocked with speed governors and train control equipment so as to apply the brakes automatically if speed restrictions are exceeded or if there is another train or open switch ahead of the train. Many other variations and refinements of control are used with modern railway air brakes.

Another example of pneumatic remote control is the pilot house controls for diesel marine engines. *Fig.* 7 illustrates such a control for small clutch reversing engines. Three push buttons pilot the Tridyne Positioner which shifts the clutch to neutral, ahead, or astern. An H-2-F Controlair valve pilots the actuator which positions the engine governor. An

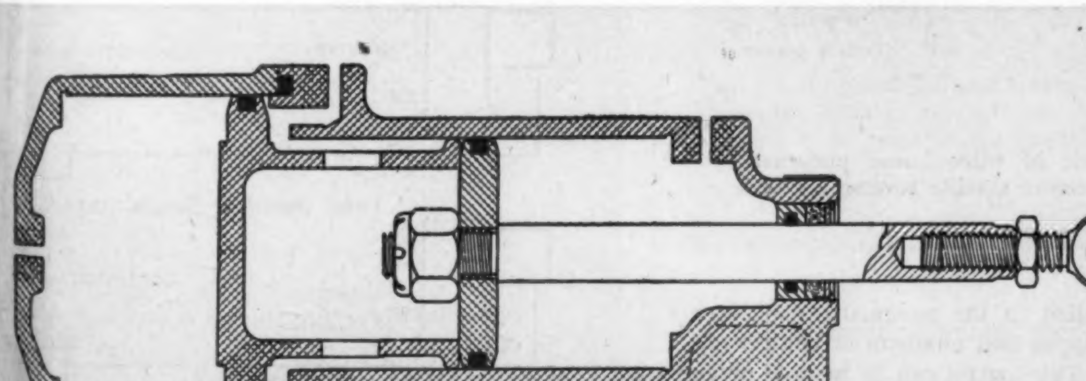


Fig. 5—Above—The Tridyne Positioner, a three-position cylinder for predetermined movements

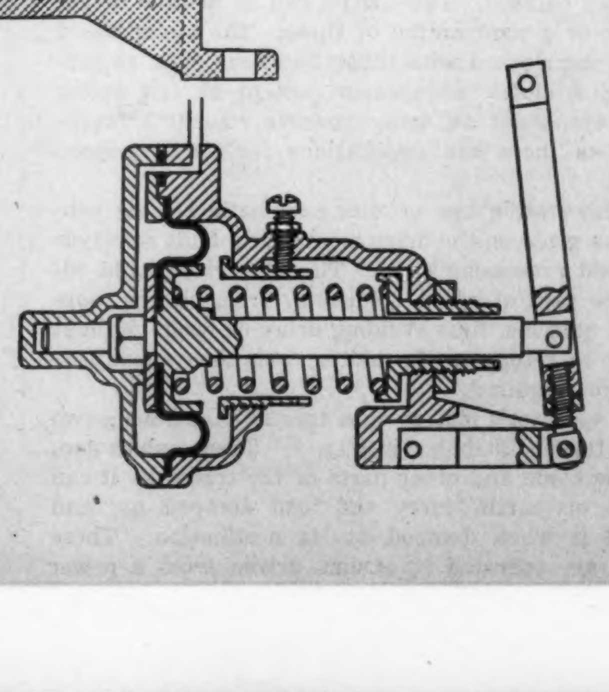


Fig. 6—Right—Simplest type of throttle actuator of the diaphragm and spring design

optional feature, shown in dotted lines, interlocks throttle and clutch controls so that the engine governor is always returned to idle position before a clutch engagement is made. Many variations of this remote control are possible such as using two or more optional control stations or controlling two or more engines from one station. Additonal interlocking and timing features can also be added where desired. Similar controls are also available for larger reversing type diesel engines.

A pneumatic actuator for operating a particular device can be piloted by many different means. For example, *Fig.* 8 illustrates a variable-speed electric motor drive on which a pneumatic positioner is mounted. The speed of the motor varies directly as the

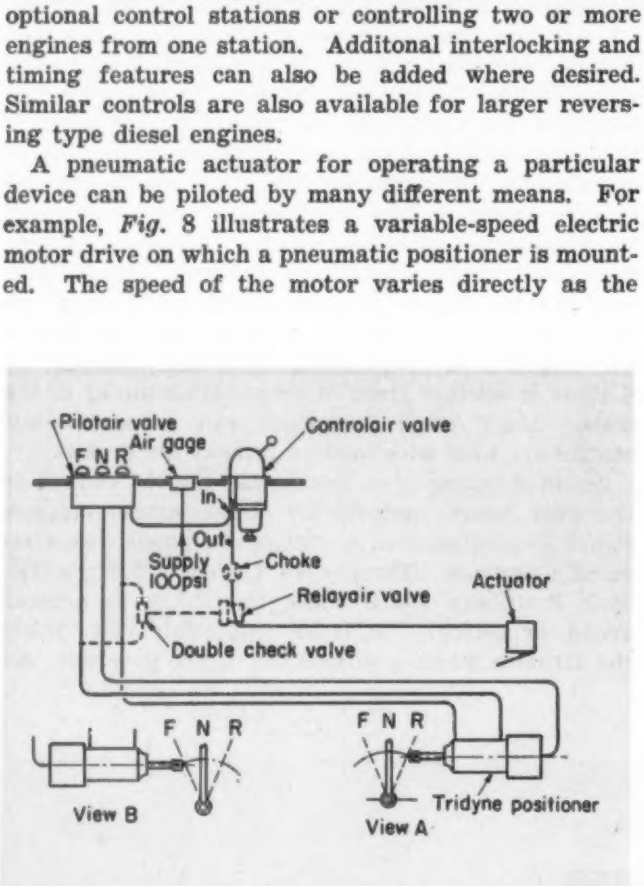


Fig. 7—Schematic of pilot house pneumatic controls for a remote marine reversing clutch

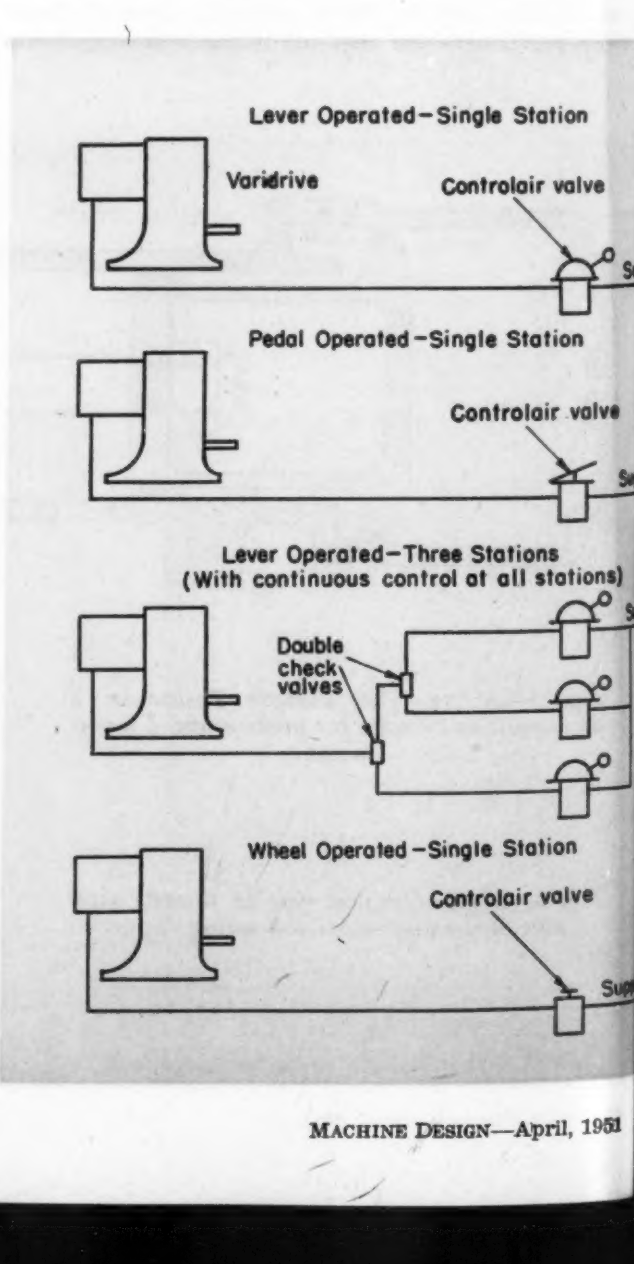
pilot pressure supplied to the pneumatic positioner. As shown various types and numbers of control stations can be used. The control can be manual or automatic or a combination of these. The motor speed can be interlocked with other functions such as preventing a clutch engagement except at low speed. There are about as many possible control arrangements as there are applications for variable-speed motors.

An interesting use of this pneumatic remote control was made on the drive motor for a fruit conveyor in a food processing plant. The supervisor could adjust the rate of speed frequently from one or more remote stations, thus avoiding delay or waste without having to return to the motor each time an adjustment was required.

The operator's station on a tractor that pulls a two wheel trailer is shown in *Fig.* 1. Three cables control the blade and other parts of the trailer so it can scrape up earth, carry the load scraped up, and spread it when dumped at its destination. These cables are operated by drums driven from a power take-off on the tractor. The driver, in addition to running the tractor, must operate these three cables and position them accurately for the work to be done. This he does with three air valves shown in the upper center of the illustration. Each valve controls a built-in clutch cylinder and a brake for one cable.

Prior to the design of pneumatic controls for a machine like this, the cable drums were controlled through mechanical linkages. These required frequent adjustment to compensate for wear, and the physical effort required of the operator was much greater. The present machine, with pneumatically controlled cable drums and air-operated vehicle brakes, can be run with little effort and correspondingly less operator fatigue.

The same kind of controls are used on power shovels, cranes, bucket dredges, and other machines where heavy hoisting drums are used. Pneumatic equipment not only permits remote control without a lot of mechanical linkages, it also permits practically effortless control of heavy machinery. This is important as it permits the less muscular operator to accomplish as much work as the big brawny fellow. Compact, easily operated controls, such as shown in *Fig.* 9, also reduce accidents and man failures in



proportion to the reduction of operator fatigue.

AIRCRAFT PNEUMATIC CONTROLS: A relatively new field for pneumatic power devices and remote control is on aircraft. Such tasks as retraction of landing gear and opening bomb doors require large-power, rapid operation for short intervals. Many other auxiliary devices need to be power operated by remote control. Until recent years only electric and hydraulic equipment were available but recent development of pneumatic devices especially designed for aircraft service shows promise of a wide field of application for pneumatic controls in the aircraft industry, *Fig.* 10.

Higher Pressures Practicable

For purposes of weight reduction and to meet space limitations, air pressures of 1000 psi to 1500 psi are currently being used and higher pressures are being considered. This permits small, compact operating cylinders to deliver power at points where space and weight limitations are practically prohibitive for other control mediums.

One of the characteristics of pneumatic equipment which makes it particularly suited for aircraft is that the stored energy in the form of high pressure air requires little weight. The air itself has an insignificant weight and can be stored in tubular structural members of the plane so that little added weight or space is needed for storage tanks.

A small engine-driven compressor can be used and advantage can be taken of a ground charging plant that will initially store full pneumatic energy before take-off. This prevents the compressor from using engine horsepower when it is needed most.

High-pressure air systems require accurate workmanship in manufacture of valves and other devices to keep leakage to a minimum. However, from a maintenance standpoint, leakage is easily detected in the system and, furthermore, does not cause fire hazard.

Use of air brakes on landing wheels of planes permits rapid brake applications or releases with maximum flexibility of control. The use of air brakes on aircraft also led to the use of Decelostat controllers. The Decelostat controller was originally developed for railway passenger cars to prevent skidding or sliding wheels when brake applications were made on rails with low adhesion, oil, frost, etc. It functions by reading the high deceleration rate that occurs

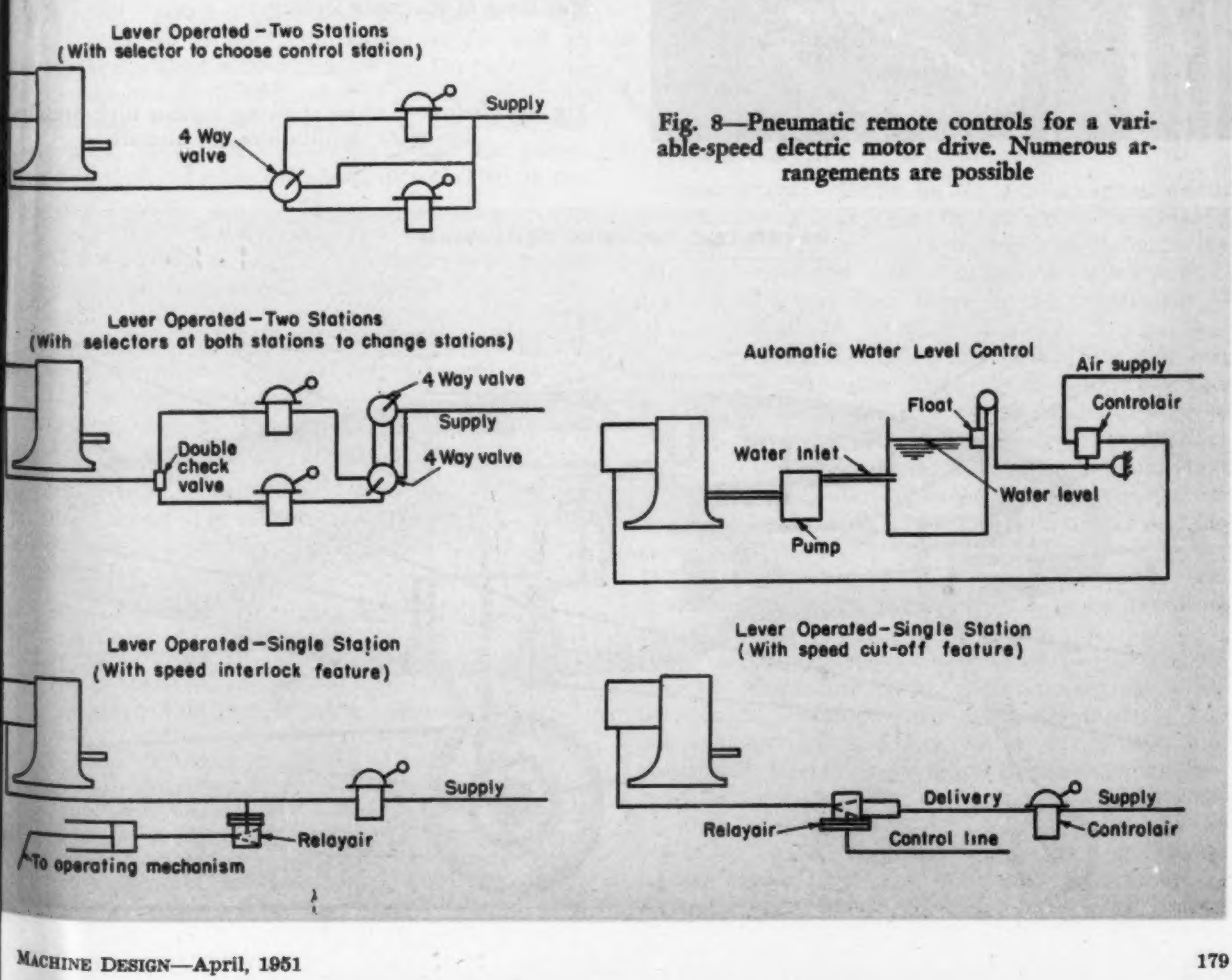


Fig. 8—Pneumatic remote controls for a variable-speed electric motor drive. Numerous arrangements are possible

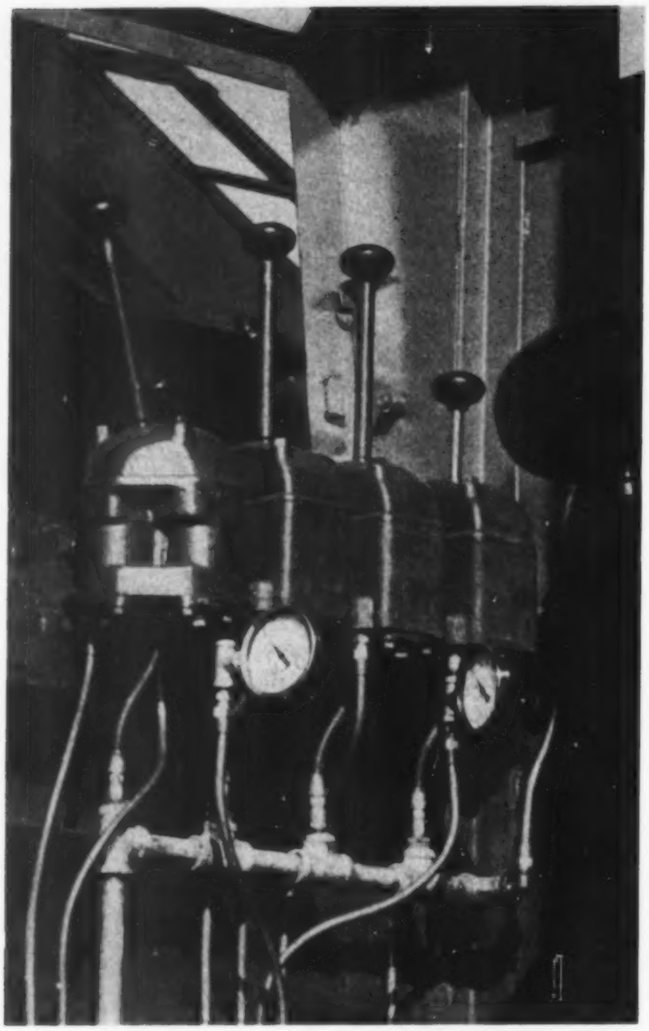


Fig. 9—Left—Pneumatic controls are compact and easily operated as indicated by this typical crane and shovel installation of four control valves

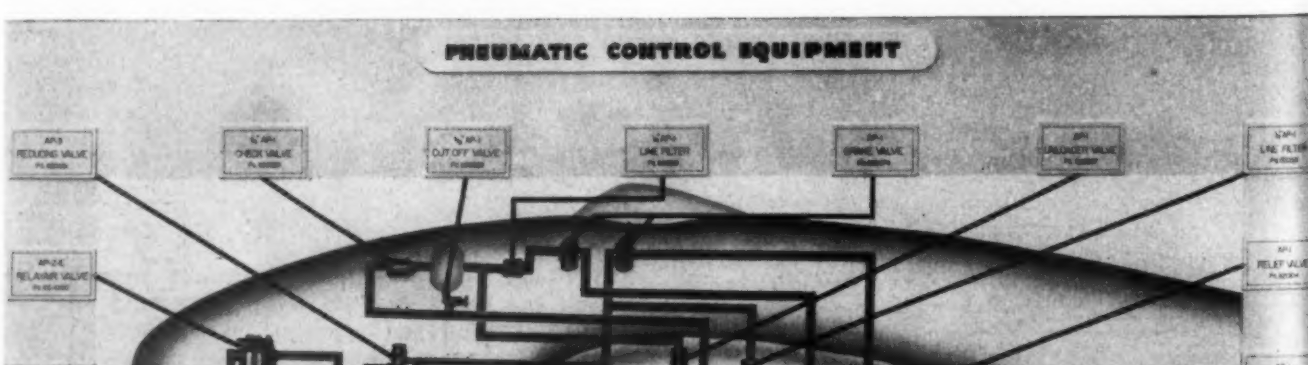
when wheels are about to slide and relieving the brakes momentarily until the wheels attain normal speed again.

The Decelostat controller for aircraft is an important aviation development because wheel sliding during airplane landings is a source of tire wear, blowouts, and unnecessary hazard to plane and personnel. On the large, high-speed, heavy planes of today these controllers permit short stops with less tire wear and hazard. This is obtained without the necessity of great skill on the part of the pilot in handling the brakes.

Broad Design Horizons

In aircraft as in many other fields the designers are finding more tasks for compresser air. Many pneumatic devices designed for a specific application are finding uses in entirely different fields. The flexibility of pneumatic equipment will lead to even wider fields of use. The problem of designing satisfactory air supply and storage systems will be discussed in the second part of this series which will appear in the May issue of MAÇHINE DESIGN.

Fig. 10—Below—Layout showing various high-pressure pneumatic applications for aircraft



A Nonreversing Coupling

. . . sensitive to velocity and acceleration

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WHEN a deep-well centrifugal pump is in normal operation, the discharge pipe extending from the pump at the bottom of the well to the surface is filled with water. When the pump is to be stopped, and its driving motor is disconnected from its source of power, the water in the discharge pipe flows back down this pipe and through the pump. This flow tends to drive the pump, as a turbine, in the reverse direction.

For a very deep well, the negative acceleration may be so great as to stop the pump and motor abruptly, then accelerate them to a high speed in the reverse direction. Unless means are provided to prevent reversal of the pump and motor, the speed attained in the reverse direction may be great enough to cause damage to the rotating parts. A description and an analysis of a type of nonreversing coupling for such applications are given in this article.

DESIRED CHARACTERISTICS: The ideal nonreversing coupling for use with deep-well centrifugal pumps and similar applications should have the following characteristics:

1. Sensitivity to speed of rotation in such a way as to maintain the device inoperative at high rotative speeds in either direction, but be free to engage and thus become operative at low and zero speeds
2. Sensitivity to angular acceleration in such a way that a negative acceleration, or deceleration, tends to cause the device to engage, or lock. In order that it will always have sufficient time to engage, this tendency should be proportional to the deceleration. If this is accomplished, the angular displacement of the rotating members during the active period of the device will be a constant, independent of the deceleration
3. Elastic means of absorbing the torsional shock of engagement
4. Positive locking action instead of friction which is undependable
5. Inexpensive construction.

The following is a description of a nonreversing coupling that meets these requirements.

DESCRIPTION: Shown in *Fig.* 1 is a vertical cross-section through the upper end of a pump motor and coupling and a horizontal cross-section through the coupling. For the sake of clarity, certain simplifying liberties have been taken in the preparation of these sketches.

As illustrated in *Fig.* 1, the rotor of the motor is mounted in the motor frame by means of bearings for rotation about the axis $O-O$. Fixed and keyed to the motor shaft are two disks held together by screws. The disks have pockets to accommodate pawls which are pivoted on the pins and spaced from the disks by washers. Surrounding the disks and the pawls is a stationary ratchet member, fixed to the frame or end-shield of the motor. This member has a number of internal ratchet teeth for engagement with the pawls.

The number of ratchet teeth may be one greater than the number of pawls, thus reducing the angle between the locking positions of the device. For example, as shown in *Fig.* 1, the device has four pawls and five ratchet teeth. Thus the angle between locking positions of this design will be $360/(4 \times 5) = 18$ degrees.

For the pawls, the terms *leading tip* and *trailing tip* are fixed by the normal direction of rotation of the rotor, here shown as counterclockwise. Pawls

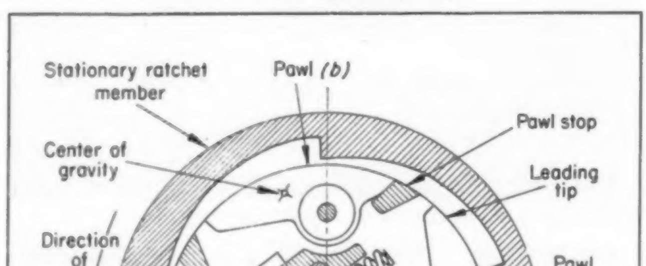
(*a*) and (*b*) are shown in their inactive positions, pawls (*c*) and (*d*) in their active or locking positions. Formed on either or both disks are projections that serve as stops to prevent the pawls from turning clockwise beyond their inactive positions.

In the disks are elastic members such as either springs, as shown adjacent to pawls (*a*) and (*d*), or blocks of rubber-like material, as shown adjacent to pawls (*b*) and (*c*). These elastic members serve to relieve the torsional shock when the pawls lock the rotor against reverse rotation. This snubbing action takes place when the leading tips of the pawls strike the elastic members.

One of the most important features of this device is that the center of gravity of a pawl, when inactive, lies at a greater radius from the center of the rotor than does the pivot axis of the pawl, and is angularly displaced in the direction of rotation from the pivot axis. This feature helps to give the device its desired functional characteristics.

OPERATION: *Fig.* 2 shows one of the pawls in its inactive position with the equivalent acceleration reaction forces A and C acting at its center of gravity.

Fig. 1—Vertical and horizontal cross-sections through nonreversing coupling



Force A is proportional to the angular deceleration of the rotor. Force C is proportional to the square of the angular velocity of the rotor.

At high rotative speeds, force C will be large compared to force A. The sum of their moments about the pivot axis will be in such a direction as to maintain the pawl in its inactive position. As the rotative speed decreases, force C decreases until finally an instant of time, $t = t_0$, is reached at which the direction of the moment of forces about the pivot axis is reversed. This instant, when $t = t_0 = 0$, will be known as the beginning of the *active period* of the pawls. At some later time, $t = t_1$, the rotor comes to rest. The interval from t_0 to t_1 will be known as the first half of the active period.

If, for the moment, it is assumed that the pawl remains in its inactive position, the rotor will then reverse and its speed in the reverse direction will increase until the force C becomes sufficiently large to again reverse the direction of moments about the pivot axis of the pawl. This occurs at a time $t = t_2$. This instant of time will be known as the end of the active period. The interval from t_1 to t_2 will be known as the second half of the active period.

As shown in the Appendix, the angular displacement of the rotor during each half of the active period is given by

$$\Delta\theta_1 = -\Delta\theta_2 = \frac{(k^2/Rr + \cos\phi_0)}{2\sin\phi_0} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

where k = radius of gyration of the pawl about its pivot axis and R, r and ϕ_0 are as shown in *Fig.* 2. The important feature of Equation 1 is that the angular displacement of the rotor during the active period of the pawls is independent of the angular acceleration of the rotor, and is fixed only by certain nondimensional design parameters of the device.

In addition to the effect of force A in activating the pawls, another significant effect is brought out in the Appendix. This effect is due to the angular velocity of the pawls which exists at the beginning of the active period. Since, during the active period, no force tends to decrease the angular velocity of the pawl while that of the rotor is decreasing, the pawl moves on out into its active position.

The question now arises as to whether there is sufficient time during the active period for the pawls to attain their locking positions. Since the complete differential equation of motion of the pawls is not directly integrable, a minimum angular displacement of the pawls during the active period may be approximated by neglecting the effect of forces A and C. This is equivalent to assuming that the pivot axis of the pawl passes through its center of gravity.

As shown in the Appendix, there is sufficient time during the first half of the active period for an angular displacement of the pawls of

$$\Delta\phi_1 = \frac{(k^2/Rr + \cos\phi_0)}{2\sin\phi_0} \quad \ldots\ldots\ldots\ldots\ldots (2)$$

Also, there is sufficient time during the entire active

period for an angular displacement of the pawls of

$$\Delta\phi_2 = \frac{2\ (k^2/Rr + \cos\phi_0)}{\sin\phi_0} \quad \ldots\ldots\ldots\ldots \text{(3)}$$

It is important to note that combining Equations 1 with 2 or 3 gives

$$\Delta\phi_1 = \Delta\theta_1 \quad \ldots\ldots\ldots\ldots \text{(4)}$$

$$\Delta\phi_2 = 4\Delta\theta_1 \quad \ldots\ldots\ldots\ldots \text{(5)}$$

These quantities are also independent of the angular acceleration of the rotor. Therefore, no matter how rapidly the rotor decelerates, there may always be sufficient time for the pawls to take up their locking positions.

As a design example, assume that such a coupling has four pawls and five ratchet teeth. Then, the angle between locking positions of the rotor will be 18 degrees. Suppose that the angular displacement of the rotor during the first half of the active period should be 15 degrees, or $\pi/12$ radian. Values of k, R, r and ϕ_0 would then be chosen so that

$$\frac{k^2/Rr + \cos\phi_0}{2\sin\phi_0} = \frac{\pi}{12} \quad \ldots\ldots\ldots\ldots \text{(6)}$$

This may be done quite readily with four variables to manipulate.

It should be noted that when a pawl first engages with one of the ratchet teeth, the force exerted on the pawl by the tooth is in such a direction as to further engage the pawl, and this force will remain small until the pawl is fully engaged. This minimizes the danger of the ratchet teeth's being chipped by a partially engaged pawl.

CONCLUSION: The preceding discussion has shown how the coupling described meets the first four requirements of such a device. With reference to the fifth requirement, a low manufacturing cost, the only really essential machine work on this design, with the exception of fitting it to the motor shaft, is the facing of the mating surfaces of the two disks, drilling and tapping for the screws, and drilling the disks and pawls for the pins. No fine clearances are required. Probably, the larger the clearances the more reliable the device will be.

The pawls may be steel or malleable castings and need not be made of stainless material if stainless pins and washers are used. Thus, there would be no moving contact between parts that might rust.

Nomenclature

R = Radius from axis of rotation of rotor to pivot axis of pawl
r = Radius of center of gravity of pawl from its pivot axis
t = Time
t_0 = Time at beginning of active period
t_1 = Time at end of first half of active period, when $\omega = \omega_1 = 0$
t_2 = Time at end of active period
α = Angular acceleration of rotor, assumed to be constant during active period
k = Radius of gyration of a pawl about its pivot axis
ϕ = Angular displacement of a pawl relative to rotor
θ = Angular displacement of rotor
$\omega = d\theta/dt$ = Angular velocity of rotor

APPENDIX: If the angular acceleration of the pawl is referred to the rotating member, it may be shown that

$$\frac{k^2}{Rr}\frac{d^2\phi}{dt^2} = -\alpha\ (k^2/Rr + \cos\phi) - \omega^2\sin\phi \quad \ldots \text{(7)}$$

The motion of the rotor is described by the equation

$$\frac{d\omega}{dt} = \alpha \quad \ldots\ldots\ldots\ldots \text{(8)}$$

These two equations are the basic ones describing the operation of the device.

Integrating Equation 8 gives

$$\omega = \alpha t + c_1 \quad \ldots\ldots\ldots\ldots \text{(9)}$$

At the beginning of the active period when $t = t_0 = 0$, $\omega = \omega_0$. Hence, $c_1 = \omega_0$ and

$$\omega = \alpha t + \omega_0 \quad \ldots\ldots\ldots\ldots \text{(10)}$$

Since

$$\omega = \frac{d\theta}{dt} \quad \ldots\ldots\ldots\ldots \text{(11)}$$

$$\frac{d\theta}{dt} = \alpha t + \omega_0 \quad \ldots\ldots\ldots\ldots \text{(12)}$$

Integrating Equation 12,

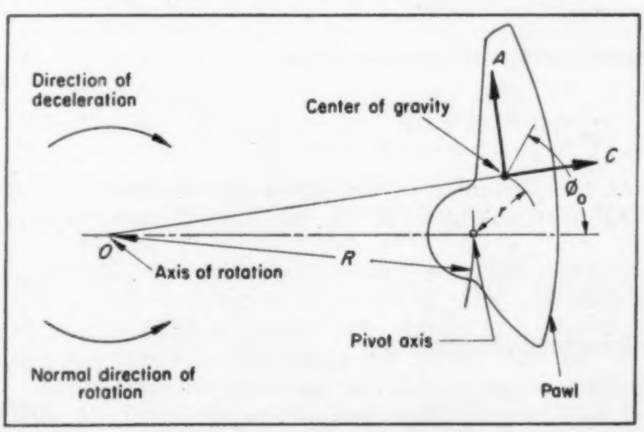
$$\theta = \frac{\alpha t^2}{2} + \omega_0 t + c_2 \quad \ldots\ldots\ldots\ldots \text{(13)}$$

Since, at the beginning of the active period when $t = 0$, $\theta = \theta_0$ and $c_2 = \theta_0$,

$$\theta = \frac{\alpha t^2}{2} + \omega_0 t + \theta_0 \quad \ldots\ldots\ldots\ldots \text{(14)}$$

At the end of the first half of the active period when

Fig. 2—Force diagram for one of the pawls shown in its inactive position, decelerating counterclockwise



$t = t_1$, $\omega = 0$. Substituting these values in Equation 10 and solving for t gives

$$t_1 = -\frac{\omega_0}{\alpha} \quad (15)$$

Substituting this in Equation 14 and solving for the angular displacement of the rotor during the first half of the active period,

$$\Delta\theta_1 = \theta_1 - \theta_0 = -\frac{\omega_0^2}{2\alpha} \quad (16)$$

Referring now to Equation 7, at the beginning of the active period when $t = t_0 = 0$, $d^2\phi/dt^2 = 0$, $\phi = \phi_0$ and $\omega = \omega_0$. Substituting these values in Equation 7 and solving for ω_0^2/α

$$\frac{\omega_0^2}{\alpha} = \frac{(k^2/Rr + \cos\phi_0)}{\sin\phi_0} \quad (17)$$

Substituting this in Equation 16 gives

$$\Delta\theta_1 = \frac{(k^2/Rr + \cos\phi_0)}{2\sin\phi_0} \quad (18)$$

Equation 17 is satisfied by either positive or negative values of the angular velocity of the rotor. Hence it can be shown that during the second half of the active period, the angular displacement of the rotor is equal to that during the first half but in the reverse direction. Thus

$$\Delta\theta_1 = -\Delta\theta_2 = \frac{(k^2/Rr + \cos\phi_0)}{2\sin\phi_0} \quad (19)$$

This is Equation 1 appearing in the preceding explanation.

In order to approximate the minimum angle through which a pawl may turn during the active period, to determine if there is sufficient time for a pawl to take up its locking position during that period, reference is made to Equation 7. Since this equation is not directly integrable, the terms containing $\cos\phi$ and $\sin\phi$ will be neglected. This is equivalent to assuming that the pivot axis of the pawl passes through its center of gravity, or that $r = 0$. Then Equation 7 becomes

$$\frac{d^2\phi}{dt^2} = -\alpha \quad (20)$$

Integrating this equation gives

$$\frac{d\phi}{dt} = -\alpha t + b_1 \quad (21)$$

At the beginning of the active period, when $t = 0$, $d\phi/dt = 0$ and hence $b_1 = 0$. Equation 21 then becomes

$$\frac{d\phi}{dt} = -\alpha t \quad (22)$$

Integrating again,

$$\phi = -\frac{\alpha t^2}{2} + b_2 \quad (23)$$

At the beginning of the active period, when $t = 0$, $\phi = \phi_0$. Hence, Equation 23 becomes

$$\phi - \phi_0 = -\frac{\alpha t^2}{2} \quad (24)$$

The angular displacement of the pawl at the end of the first half of the active period is then found by substituting $t = t_1$ from Equation 15, or

$$\Delta\phi_1 = \phi_1 - \phi_0 = -\frac{\omega_0^2}{2\alpha} \quad (25)$$

or, substituting Equation 17,

$$\Delta\phi_1 = \frac{(k^2/Rr + \cos\phi_0)}{2\sin\phi_0} \quad (26)$$

This is Equation 2 appearing in the earlier discussion.

Similarly, the possible minimum angular displacement of a pawl during the entire active period is found to be

$$\Delta\phi_2 = \phi_2 - \phi_0 = -\frac{2\omega_0^2}{\alpha} \quad (27)$$

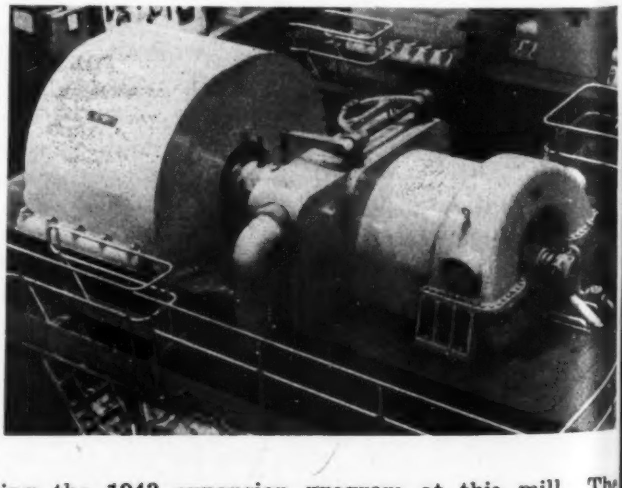
or, substituting Equation 17,

$$\Delta\phi_2 = \frac{2\,(k^2/Rr + \cos\phi_0)}{\sin\phi_0} \quad (28)$$

This is Equation 3 appearing in the preceding discussion.

New Blower for Steel Works

LARGE blast furnace turbo blower, recently installed by Dravo Corp. Machinery Division at the Edgar Thomson Works of Carnegie-Illinois Steel Corp., has joined two others placed in operation during the 1943 expansion program at this mill. The new De Laval machine has a capacity of 121,000 cubic feet of blast air per minute while the other two can each supply 90,000. The turbo blowers are designed for 650 psi steam and 750 F at the throttle.



By Dietrich W. Botstiber
Chief Mechanical Development Engineer
Piasecki Helicopter Corp.
Morton, Pa.

Helicopter Drives

. . . mechanical transmission system combines light weight with high safety factor

HAVING passed the "Kitty Hawk" stage, the modern transport helicopter is much more than just a strange contraption. Its performance is so unique in almost completely eliminating the limitations of terrain and space that its place in the transportation system of the future is beyond any doubt. Now the engineers are faced with the task of shortening the span between today and the time when helicopters are common carriers as railroads, automobiles, or fixed-wing aircraft.

Mechanical transmission of power is used in the helicopter to a far greater extent than on other types of aircraft. There is, therefore, a wide field for drive design and development work in this industry. Requirements for weight reduction, operating safety, and operating efficiency, become highly acute on items that on other types of equipment are of minor importance.

Prior to a consideration of drive problems, a brief review of the characteristics of the helicopter itself is pertinent. The idea of a rotating wing is not new. When it was first considered, centuries ago, power limitations prevented its development. Later, in the twentieth century, engines of sufficiently light weight and adequate power became available, but new limitations arose because of the difficulties encountered in controlling the aircraft.

In 1936 the Focke-Wulf helicopter was flown successfully in Germany. It used cyclic pitch control for its two rotors which were arranged on the ends of booms at each side of the fuselage, *Fig. 1a*. This two-rotor arrangement eliminated the need for a special device for counteracting the rotor torque, since the two rotors turned in opposite directions.

In 1941, Platt LePage completed a similar type of helicopter in this country.

In 1939, Sikorsky flew his first successful helicopter. It also used cyclic pitch control, a separate rotor of small size being arranged at the tail for conteracing the torque of the single lift rotor, *Fig. 1b*.

During the war, Flettner in Germany experimented with laterally disposed intermeshing rotors, *Fig. 1c*

and in 1944 the Kellett helicopter, based on principles similar to the Flettner ship, was completed and flown. In the same year Hiller produced the first successful American helicopter with two coaxial lift rotors, *Fig.* 1*d*.

Early in 1945, Piasecki flew the first tandem helicopter, *Fig.* 1*e*, marking a new era in rotating wing aircraft development. Piasecki, in 1943, had completed and flown a single-rotor ship that served for studies and development and finally led to the conception of the tandem principle, which since then has proved its outstanding advantages.

Design of a helicopter may be divided into six basic groups: Fuselage, rotor system, control system, equipment, power plant, and power transmission.

Fuselage, equipment, and power plant are of a nature comparable to those for fixed wing aircraft although the different flight conditions of the helicopter have a marked effect on their design, and thus produce essential deviations in even these groups from conventional aircraft arrangements.

Control system of the helicopter bears little resemblance to that of fixed-wing aircraft. This is due to the difference in flight conditions and also to the fact that the lift system of the helicopter is subjected to dynamic and fluctuating loads—in contrast to the more static nature of the fixed wing—leading to different design conditions.

Replacing the wing of conventional aircraft, the rotor of the helicopter has little in common with a conventional aircraft wing except for certain basic aerodynamic characteristics.

A Power Transmission Unique to Aircraft

The power transmission system, a component usually not found in conventional aircraft, is as characteristic of the helicopter as the lift rotor and its controls. Development of the transmission system, however, followed a different line, since its design originally could be based on principles evolved a long time previously and developed and tested under similar, though not equally severe, operating requirements.

Engine power must be transmitted to the rotors, with a reduction in speed. Also, the engine must be started idle — disconnected from the rotors — after which the rotors must be brought to full speed in a gradually accelerating process. In flight the rotors must be free to rotate if the engine speed is appreciably reduced or if the engine fails. The transmission system therefore contains several basic components, which perform these functions; these components may be designed into one or more combined units.

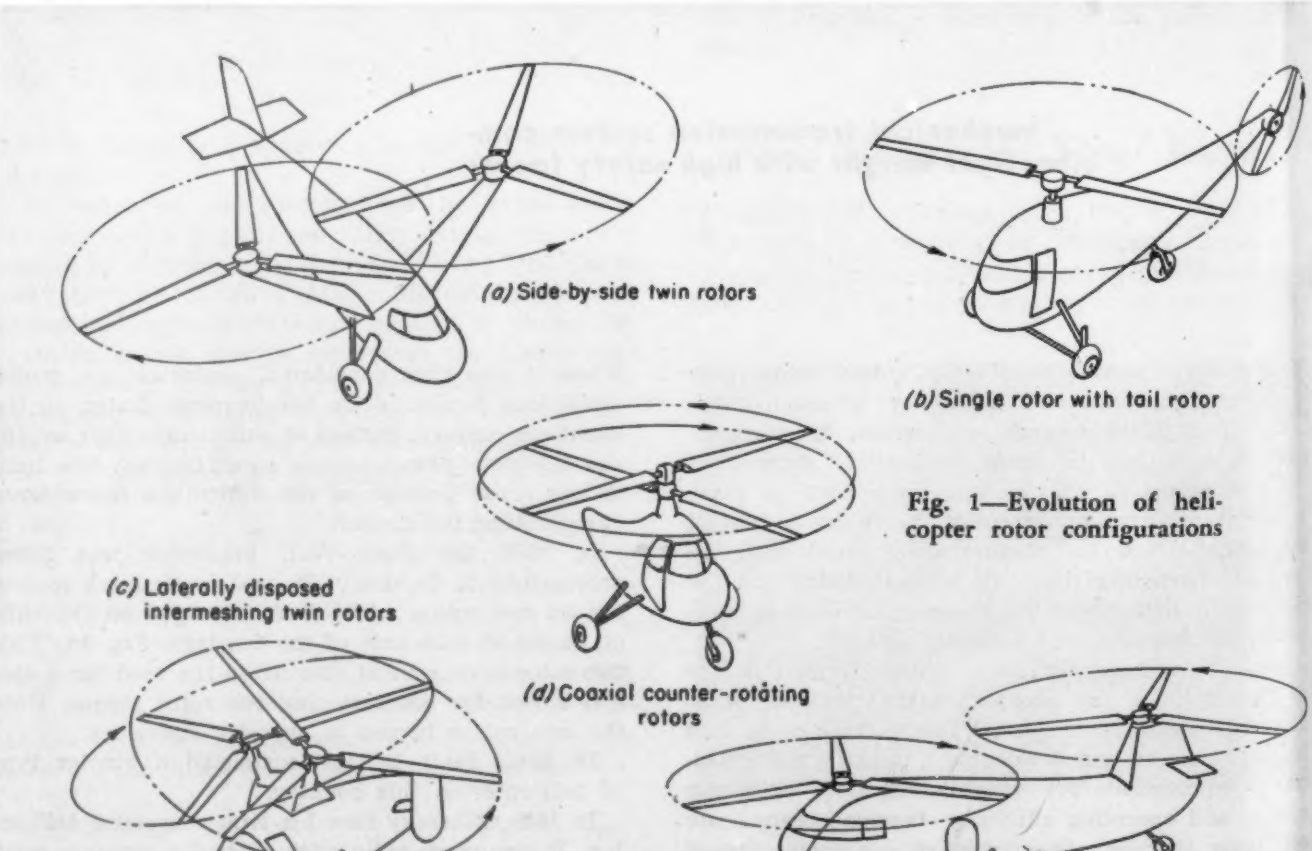


Fig. 1—Evolution of helicopter rotor configurations

On all existing helicopters the transmission system is designed with purely mechanical components, using gears for speed reduction or change in shaft direction, friction clutches for the gradual starting of the rotors, and conventional type free-wheeling units to permit autorotation of the lift rotors. Hydraulic couplings and torque converters so far have not warranted the weight increase that would result from their application. Variable speed ratios, such as provided by a hydraulic torque converter, could favorably affect engine efficiency, could improve flight performance in certain specific cases, and could reduce wear of the engine. However, the additional weight, and the power losses in the converter, offset these possible advantages.

Typical Helicopter Rotor Drives

Schematic pictures of typical helicopter drives are shown in *Figs.* 2 to 6. The single rotor drive with vertical engine represents a simple solution for small helicopters, *Fig.* 2. For larger single rotor arrangements the engine is usually set horizontally in order to obtain a suitable center of gravity location and to avoid obstruction of cabin space, *Fig.* 3. Two-rotor configurations with either one or two engines, such as the laterally displaced intermeshing configuration in *Fig.* 4, or the single and two engine tandem configuration in *Figs.* 5 and 6, are generally applied on large-size ships.

All these arrangements show the basic components of the helicopter drive: friction clutch for starting, overrunning clutch for autorotation, gearing for speed reduction and change of direction, and shafting for connection between units. Individual design of these units, and their relative arrangement within the drive system, are governed by the operating conditions of the rotor configuration used and the characteristics of rotors and power plant. By suitable arrangement of the starting and overrunning clutches in the system it is possible to start the helicopter with either one of two engines, to continue flight with only one engine, or to change to autorotation in case of engine failure.

Friction clutches used on helicopters may be (1) centrifugal, with pressure between the friction members depending on the speed of rotating parts, or (2) constant torque, with constant predetermined pressure applied by an actuator controlled by the pilot. Each system has its own operating characteristics which may be used to advantage for specific applications. The friction surfaces are usually sintered metallic material running against steel.

Several principles are employed in design of over-

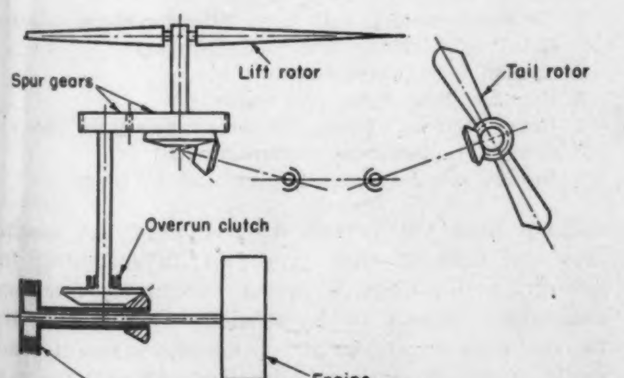


Fig. 2—Above—Single-rotor drive with vertical engine

Fig. 3—Below—Single-rotor drive with horizontal engine

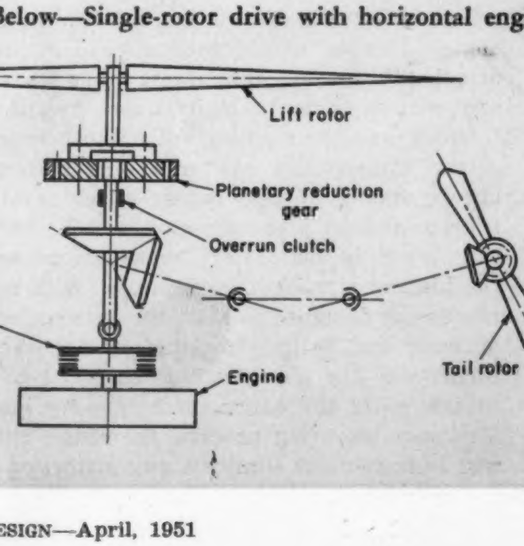
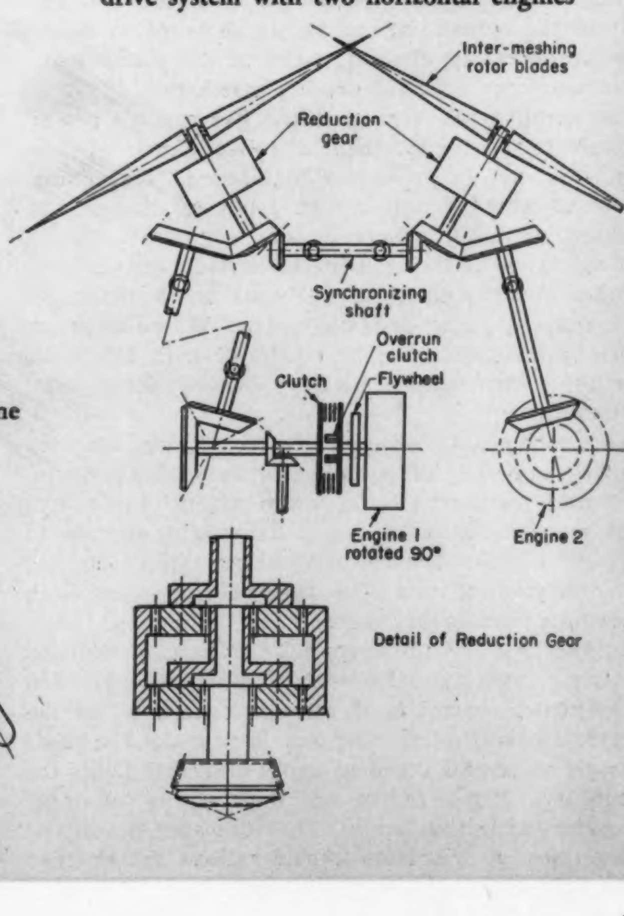


Fig. 4—Twin laterally disposed rotor drive system with two horizontal engines



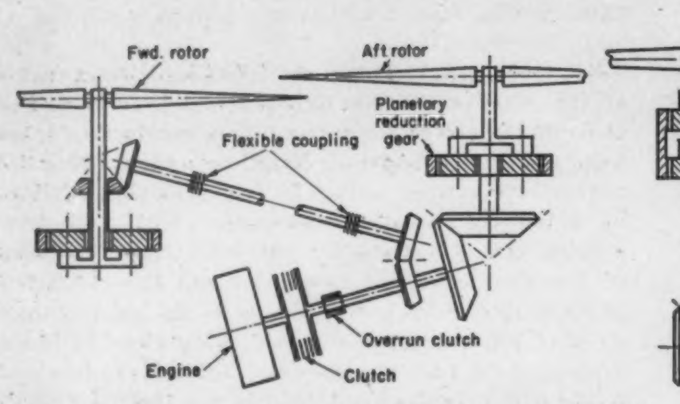


Fig. 5—Twin tandem rotor drive with single engine

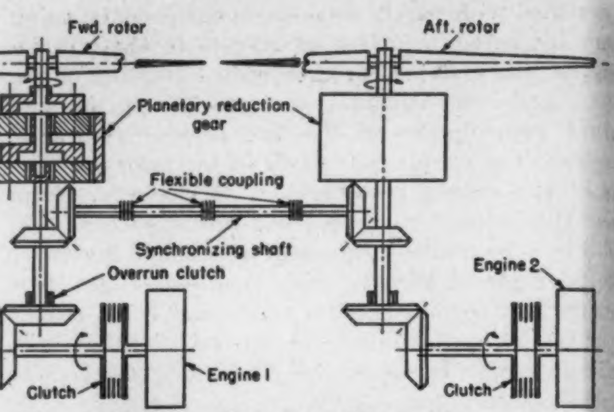


Fig. 6—Twin tandem rotor drive with two engines

run clutches, the most familiar being the roller type, which has been used extensively on automotive drives. The roller type is considered best suitable for long periods of overrunning at high speeds, such as when a two-engine ship flies on only one engine, or when a ship is being towed with its engine idling or stopped. Capacity of the roller type overrun clutch is limited by the surface stress of the rollers on the circular races. This surface stress is a function of transmitted torque, working radius of the clutch, roller diameter, and race diameter and length of contact. The operating speed also increases this stress, because of centrifugal force on the rollers. The advantage of this type overrun clutch is the uniform wear of the cylindrical rollers, which tends to maintain the operating characteristics of the clutch even when wear occurs on the driving members.

The sprag type overrun clutch has greater power transmitting capacity than a roller clutch of the same size, but in prolonged high-speed overrunning the local wear of the sprags tends to change the working surfaces, adversely affecting the capacity and safety of this clutch in the driving condition.

Other overrun clutches are based on the principle of a self-energizing friction clutch. Where such an overrunning clutch can be combined with the main starting friction clutch, a highly efficient design may be obtained.

The spring type overrun clutch, which operates on the principle of a self-energizing brake band wrapped around the brake drum several turns, has good mechanical qualities but its weight appears to be high compared with most other types. So far, no application of this type overrun clutch has been made on a production helicopter.

In shafting and universal joint design, in addition to torque capacity and operating safety, the question of vibration control is of serious concern to the designer. Generally, the tendency is to make the shaft as rigid as possible, and to apply universal joints for flexibility. Rigid shafts are dural tubes of large diameter with thin walls. The diameter usually is determined by vibration considerations rather than by stress requirements, the wall thickness being a matter of resistance against buckling plus sufficient ruggedness for handling and service.

Some universal joints provide full uniformity of speed regardless of break angle, others possess inherent cyclic speed changes as a function of the break angle. Depending on the accuracy of alignment and on the rigidity of the supporting structure, the one or the other type will be selected. The choice may be made from any of these types:

1. Constant-velocity ball joint (Bendix-Weiss, Gear Grinding Machine Company-Rzeppa)
2. Cardan (Mechanics, Spicer)
3. Flexible disk (Thomas, Hardy)
4. Gear (Barcus, Poole, Waldron, Sier-Bath, etc.)
5. Crutch (automotive manufacturers)
6. Special designs, both foreign and U. S. A.

Space does not permit discussion of the specific pros and cons of each type and its suitability for different applications. However, certain essential considerations should be mentioned. Only where the transmission arrangement contains an inherent break angle of major size (several degrees) will the constant-speed feature of some types of joints be of material advantage. Usually the shafting can be arranged so that the cyclic speed variations are canceled out by use of two joints with the same break angle. Design considerations should be directed principally toward elimination of wear and friction and with a view to maintenance requirements.

Gears and gear cases are a most important part of the transmission system of any helicopter. Gearing is mainly of two types, spiral bevel gears and spur or helical planetary units. All power-transmitting gears in helicopter transmissions are made to the highest standards of accuracy, with metallurgical processing designed to meet the extreme requirements of wear and fatigue resistance. Accuracy and uniformity of the gear teeth is essential to insure full utilization of the entire tooth face for load carrying. The manufacturing process, therefore, must be planned to prevent or eliminate any distortion of the gear

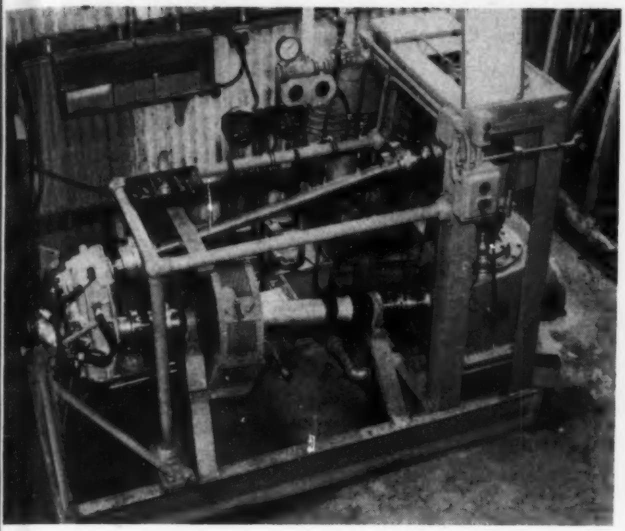


Fig. 7—Left—Test stand for helicopter transmission cases, employing locked-in torque loading principle

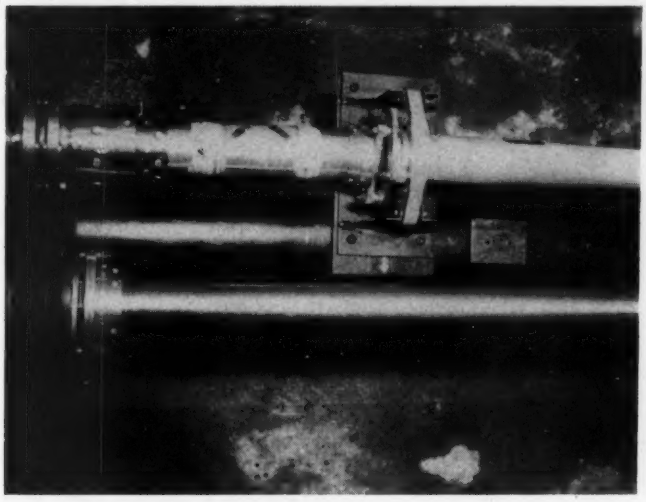
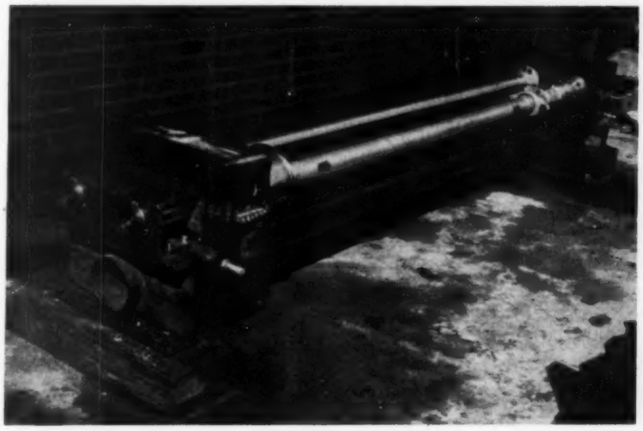


Fig. 8—Above—Test stand for shafts and couplings, with locked-in torque

Fig. 9—Below—Shaft and coupling test arrangement, using strain gages to measure twist in shaft



in heat treating. Distortion may be reduced by nitriding, in which the temperature is sufficiently low. A safer way to obtain maximum strength and accuracy is to grind the teeth after all heat treating has been completed. Extensive Stresscoat tests have proved that the uniformity of load distribution is vastly superior on a ground gear. The surface compression obtained in nitriding and case hardening increases the fatigue resistance of the gear by superimposition of a compressive stress, which reduces the effect of the cyclic load variation. A similar effect is obtained by shot peening.

Requirements for weight reduction and operating safety on helicopter transmissions are increased to an extent unknown with any other type of machinery. Generally, a helicopter transmission will have about one-twentieth the weight of an industrial transmission of equal power, speed and gear ratio. Its useful service life will be less than that of an industrial transmission, but during the pre-estimated life period it must perform with an incomparably higher degree of safety. This conflict beween two opposing requirements — weight reduction and operating safety — makes it necessary to apply the most accurate available methods of stress analysis and development testing.

An inherent advantage of the helicopter is that a major portion of its testing may be done on the ground or hovering at low altitude. As far as the drive system is concerned, such tests facilitate thorough studies of vibration criteria, load fluctuations, behavior of mounting, and installation of the drive system units with respect to each other and to other components of the ship.

Gear cases may be tested for certain functions by running at normal speed without load. Such a test proves proper functioning of the lubrication system, oil seals, breathers, and auxiliary drives. For testing the load-carrying capacity and endurance of a gear case, these major methods are available:

1. Testing in the helicopter, in flight or hovering
2. Testing in the helicopter tied to the ground
3. Testing by a special test stand simulating the rotor loads on the ground
4. Dynamometer testing
5. "Back-to-back" testing with locked-in torque.

The first two methods are very expensive and not applicable for accelerated testing at increased load. The third method is more economical and may be used with overload, but still represents an expensive procedure. Dynamometer tests are valuable for highly accurate laboratory test set-ups, but at high power rating and for continuous production testing the cost is prohibitive.

Most economical method for applying full operating load on a gear case at full speed is the "back-to-back" method. In this arrangement the power from the gear case on test is carried over another gear case with the reciprocal gear ratio back into the drive. The drive motor provides only enough power to overcome the losses in the gear case on test and in the test-stand mechanism. This arrangement may

be built up by two gear cases of the same type, one of which receives the load in the opposite direction. The basic idea of this arrangement has been used for many years in testing electrical machinery. In such applications, one of the machines is used as a generator and one as a motor. The automotive industry also has adopted the principle for production testing of gear cases.

In practical applications it usually proves more advantageous to build a test stand which only takes one gear case, instead of using two of identical type with one acting in reverse. A gear arrangement of opposite gear ratio, opposite direction of rotation, and opposite shaft angle is built into the test stand unit. Torque load is applied by a device which loads the entire mechanism within itself.

Torque-Loading Test Methods

Several methods of applying this torque load and for measuring its value have been devised. One method is based on the principle of a differential gear having a stationary planetary gear which is rotated by a certain amount in order to apply a direct load on the drive shafts which carry the sun pinions. The torque may then be measured by strain gages attached to one of the shafts in the circuit and transmitted to an indicating instrument through slip rings. Another torque-loading device consists of a pair of helical gears, one of which is axially moved, thus producing a relative rotation to the other gear. The amount of the torque is measured by hydraulic pressure which is used to perform the axial displacement of the gear. *Fig.* 7 shows a test stand used for helicopter transmission testing. This device uses a fully mechanical arrangement for applying the torque load on the gear case. The torque value is indicated by measuring the deflection of a shaft within the circuit by a simple leverage ratio. Deflection of the shaft is amplified and shows the torque in inch-pounds directly on a dial. This type of load test stand is economical and provides a simple means of applying a torque load for the patterning procedure of spiral bevel gears, and for applying static loads on the completely assembled gear for stress and deflection tests.

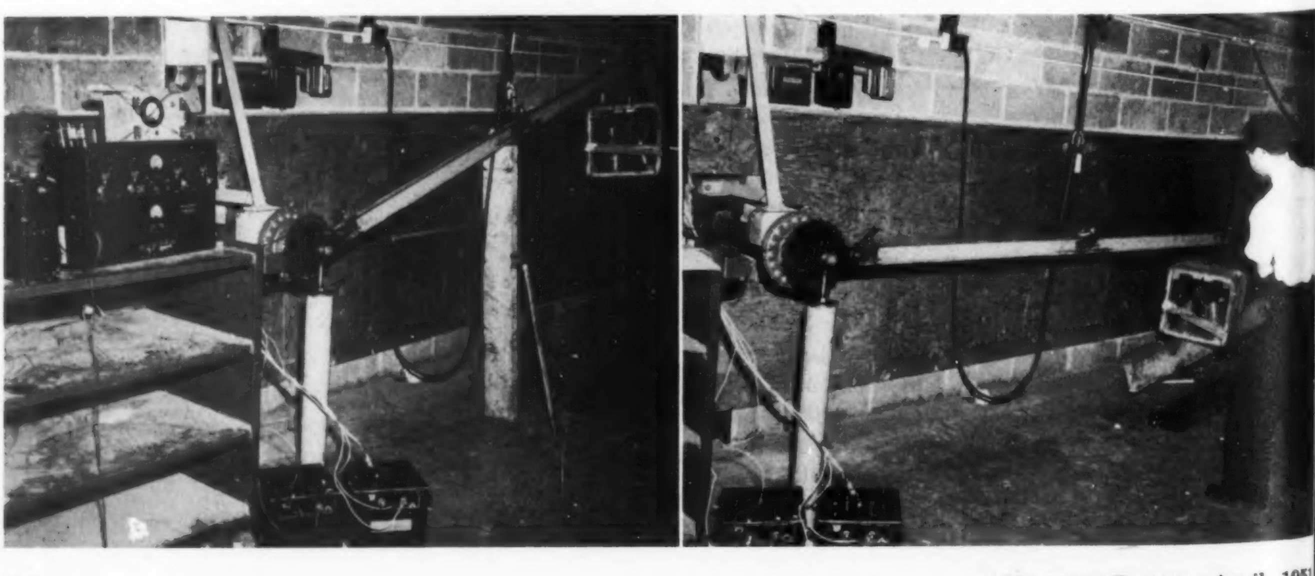
Stress and deflection tests by application of a static load on an assembled gear case are a major activity. Information on misalignment of gears and bearings under load provides data for design improvements. Deflections produced by these static loadings may be measured by various methods, the simplest one being the application of dial indicators to the points where deflections are expected. Further methods are the use of electric strain gages or brittle lacquer applications.

In addition to the tests performed on complete gear case assemblies, most individual transmission components must undergo thorough testing before application in the helicopter. Some of these tests are destructive and are carried out in order to find the ultimate load capacity of the part involved. Tests must be planned in accordance with the type of load which the part will receive in actual service. *Figs.* 8 and 9 show portions of a test stand used for endurance testing transmission shafts and flexible joints. This test stand works by the locked-in torque method. Five hundred hours of running are required at carefully calculated overloads in order to produce a fatigue failure within a reasonable time, and from the data given by these tests the actual service life of the parts under flight conditions may reliably be calculated.

Shock Load Shaft Testing

Another test stand which was used for applying shock loads to a shafting assembly is shown in *Fig.* 10. A shock load is applied by dropping weights of calculated value from a certain height. The resulting shock load, and the stresses indicated by the strain

Fig. 10—Test stand for shock-load testing of shafts and couplings: (a) prior to and (b) during shock-load application



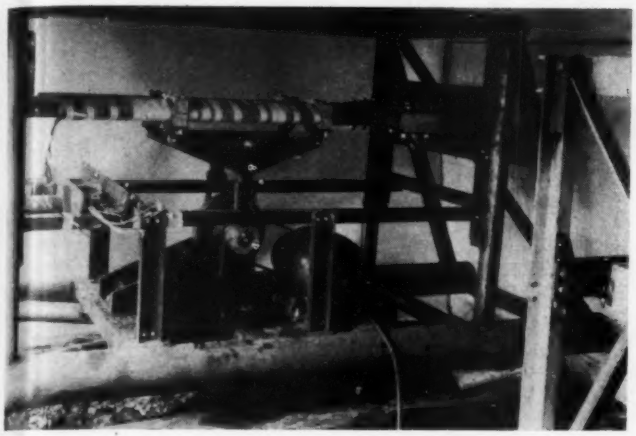


Fig. 11—Fatgue test stand for rotor blade elements

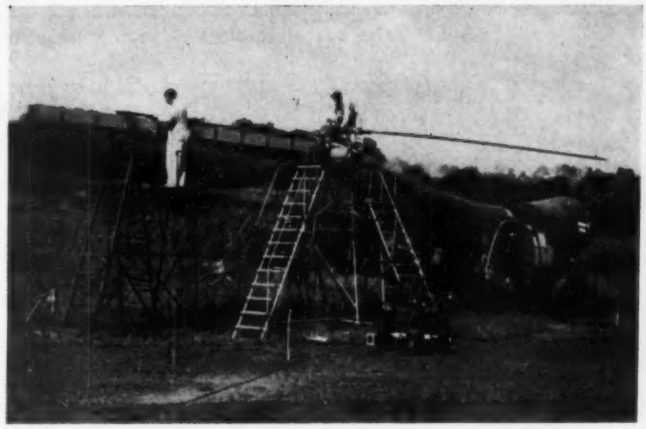


Fig. 12—Setup for testing complete rotor blades

gages, are then evaluated to calculate the capacity of the shaft under specific overload conditions that may occur in flight.

A test stand for endurance testing rotor blade elements is shown in *Fig.* 11. Here a static tension load is applied while an oscillating bending load is superimposed. In this way the bonding of the wooden blade structure to the steel spar is tested in fatigue loads equivalent to those encountered in flight. The complete rotor blade is tested on the ground by an arrangement shown in *Fig.* 12, and also in flight. The blade is mounted on the rotor hub, and supported during calibration of the strain gages by a beam which relieves the blade of all stresses; the rotor is then operated on the ground at full speed with cyclically changing control settings. A stress oscillograph is obtained by such tests, and a similar oscillograph is taken in flight, revealing the stresses in the blade during severe actual flight maneuvers.

The entire field of testing transmissions and transmission components, and the evaluation of test results for application to design data forms an extensive field which is of particular concern for the design of helicopter drive systems.

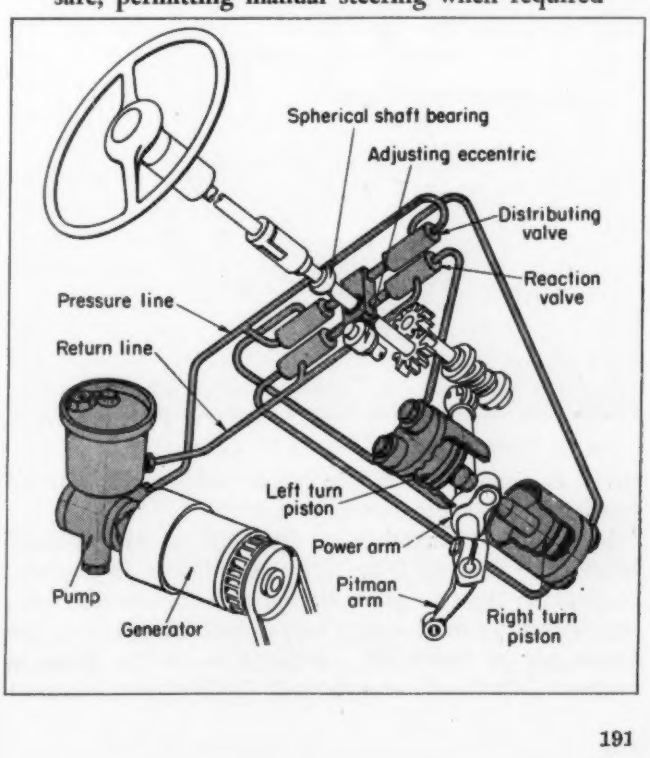
Hydraulic Power Steering for Cars

POWER steering, while not new in heavy automotive equipment, has not previously been applied to modern high-speed cars. The Hydraguide power steering mechanism, made by Gemmer Mfg. Co. and currently being installed on some Chrysler cars, requires 40 per cent less steering wheel movement and 75 to 80 per cent less effort to turn the wheels. Power control is instantaneous in response to steering wheel movement and when the wheel is released after a turn, the car recovers normally as with conventional steering gear.

The Hydraguide consists of two basic units: (1) The standard worm and roller mechanical gear with a hydraulic power device and valves built into the one housing and (2) an engine-driven Eaton hydraulic oil pump combined with an oil reservoir and filter. When the engine is not running the car steers in the conventional manner; hence, it can be steered safely if the power should fail.

A flexible fork type coupling, cushioned in synthetic rubber, joins the steering wheel shaft and its extension in the steering gear housing, *Fig.* 1. The extension shaft is mounted in a spherical bearing so its lower gear end can float several thousandths of an inch laterally to operate the valves that control the application of power. Lower end of this floating

Fig. 1—Sketch showing relationship of various elements of Hydraguide hydraulic power steering. System fails safe, permitting manual steering when required



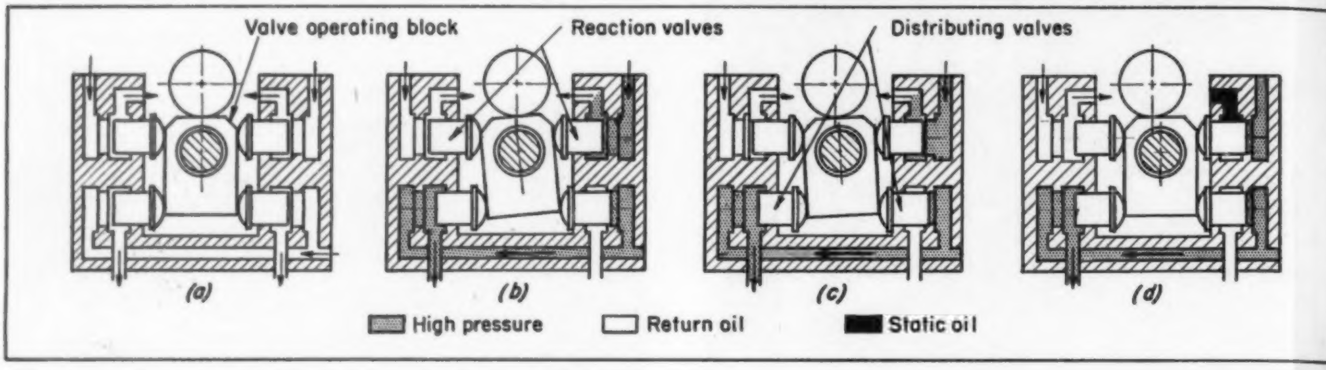
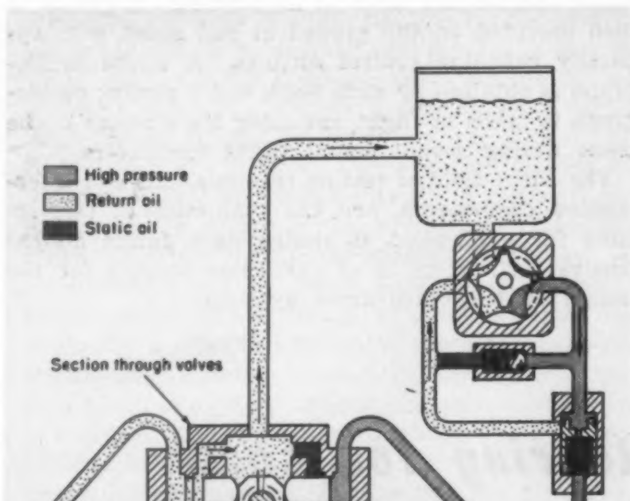


Fig. 2—Above—Simplified valve operating diagram. Valve block is shown in positions for neutral, *a,* full distributor, *b,* partial power, *c,* and full power, *d*

Fig. 3—Below—Schematic oil flow diagram of hydraulic steering unit in full power position



shaft is connected to the worm through a pair of gears having elliptoid-form teeth.

Mechanical resistance to turning of the wheels causes the floating gear to move sidewise in a plane tangent to the two pitch circles and operate the control valves by means of a valve block attached at the upper end of the shaft. Separation of the gears is prevented by an eccentrically adjustable back-up roller behind the block. Side movement of the control block may be in either direction, depending on which way the steering wheel is turned. Only 0.003-inch movement of the valves is necessary to start application of hydraulic pressure to the cylinders which power the steering gear. Application of steering effort from the hydraulic cylinders is by means of an auxiliary power arm attached to the pitman shaft between the roller gear and the pitman arm. To allow for accurate movement of the power arm, a hardened roller, needle-bearing mounted, is provided.

Valve Block Moves Pistons

Two valves are used for each power cylinder, *Fig.* 2. One—the distributor—directs oil to the proper cylinder and the other—a reactor—controls the oil flow and pressure and regulates the ratio between manual and hydraulic torque. The distributor valve must be fully displaced before the reactor valve moves. Each of the valve pistons is provided with a collar which bottoms on the valve sleeves to limit its maximum movement. The distributing valves are connected through an internal passage and, with the valve block interposed between them, are balanced so as to slide freely with the slightest movement of the valve block.

The first movement of the steering shaft swings the valve block and moves the distributor piston fully, *Fig. 2b*. Further displacement of the gear acts on the unrestrained end of the block to move the reaction piston in the same direction, *Fig. 2c* and *d*. The oil flow diagram for the power position, *Fig.* 3, shows the relationship of the power cylinders, valve operating gears, valves, oil pump, reservoir, high-pressure relief valve, and flow-control valve. The latter is a spring-loaded piston valve with an orifice to insure delivery of a uniform quantity of oil to the steering unit irrespective of variations in engine speed. This reduces the load on the pump from 60 to 75 per cent, saving power at high engine speeds.

With this design, it is possible to use a much lower overall steering gear ratio without sacrificing easy handling at low speeds or when parking. The first design will have a ratio of about 16 to 1 compared to a 26 to 1 ratio now in common use with low-pressure tires. Experimental cars are being driven successfully and safely with as low as 9 to 1 gear ratios.

Design of Logging Winch Controls

By Paul L. Brainard

Supervising Engineer, Standards Div.
Hyster Co., Portland, Oreg.

WITH the advent of large track-type logging machinery, tractor-mounted logging winches were developed which were smaller in overall dimensions than anything used previously in the woods. Tractor horsepower has approximately doubled in the last ten or twelve years with no practical increase in tractor size. As a result, clutches and brakes of tractor-mounted equipment remain essentially the same size as early models, but must handle greatly increased line pulls. This necessitated handling gear refinements and in some cases made the use of self-energizing handling gear mandatory.

The free-spooling planetary winch, *Fig.* 1, mounts on the rear face of a track-type tractor and is used for light logging and general skidding work. In operation, the drum shaft and the sun gear, *Fig.* 2, are driven through a gear reduction set by the power-take-off shaft of the tractor. The planetary gear set serves as a clutch and as a further gear reduction. The two operating drums are side by side and one of the two drums must rotate whenever the tractor master clutch is engaged. When the clutch drum, which is part of the planet cage, is restrained from rotating by a contracting brake band, the planet pinions act as idlers between the sun gear and the internal drum gear, rotating the cable drum.

For general use, a two-lever hydraulic control system has been devised with one lever for each drum. For special cases the need arose for a single-lever interlocked control which would have the following positions:

1. Neutral—brake on and clutch off.
2. Free Spooling—brake off and clutch off.
3. Spooling in—brake off and clutch on.

With heavy line pulls and the accompanying loose end band loads encountered, coupled with lever travel limited both by the "comfort zone" for the operator and physical limits imposed by the structure of the tractor itself, the design of such a single-lever control was recognized as a difficult problem. A handling gear set was designed with the clutch operating lever pivoted about the brake spring. Using this system, the same operator effort released the brake and applied the clutch. Because of limited lever travel, the force required by the operator to release the brake for free spooling (by compressing the heavy brake spring) was too great for comfort. An effort to reduce this force resulted in too little band clearance and created a drag in the free spooling position which could not be tolerated.

The handling gear was then redesigned with a conventional contracting band on the clutch drum. The brake rigging, however, consists of two bands

Fig. 1—Above—Planetary logging winch shown mounted on rear face of tractor. This equipment is used for light logging and general skidding work

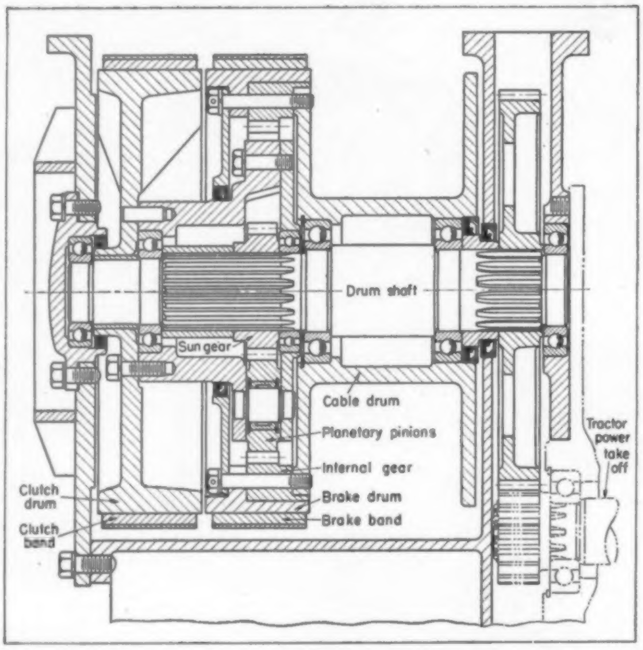
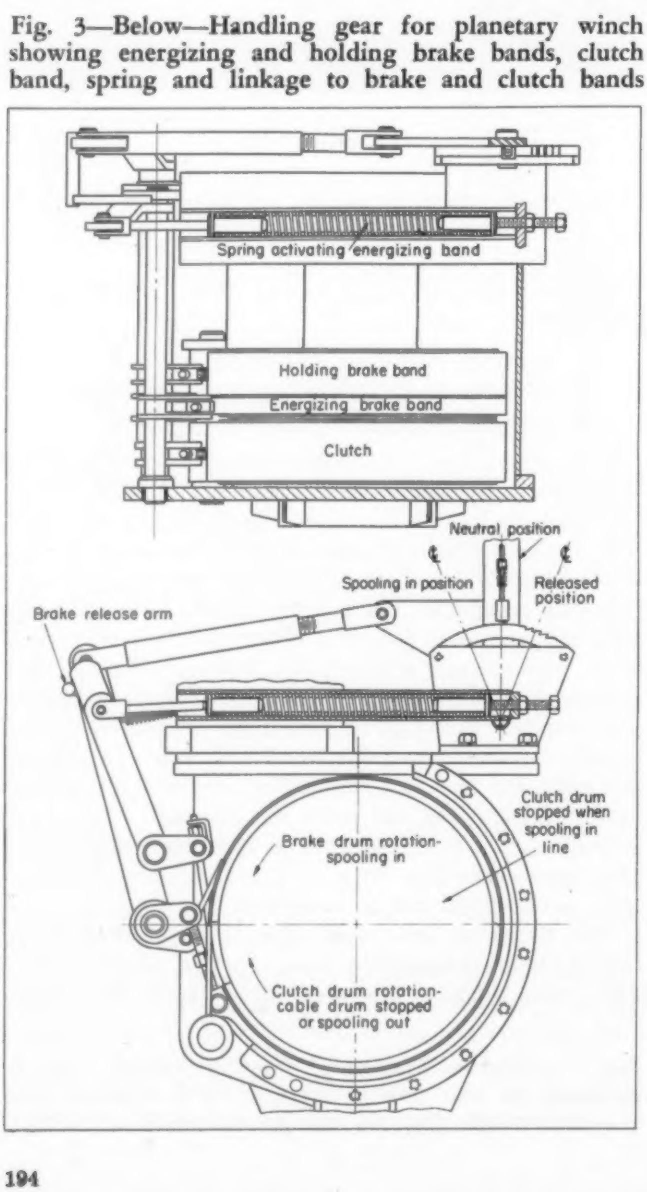


Fig. 2—Above—Cross section drawing through winch showing power take-off drive and planetary gearing

Fig. 3—Below—Handling gear for planetary winch showing energizing and holding brake bands, clutch band, spring and linkage to brake and clutch bands



on the same drum, a holding band and an energizing band, *Fig.* 3. The energizing band is narrow and is actuated by a spring of low rate. The tight end of the energizing band is connected by means of a crank to the loose end of the holding band. The only connection between the brake rigging and the hand lever is through the brake release arm by which the brake is loosened for free spooling. During the spooling-in cycle, the brake drum is permitted to drag against the loose ends of the bands.

Continuous Operation Possible

It was realized that this design did not fulfill the original specification for clutch-on—brake-off position. However, the drag, being against the loose ends, would tend to further loosen the bands resulting in an unusually small amount of heat being generated. It was felt that this small amount of heat coupled with good heat dissipation would permit continuous operation of the winch without developing excessive temperatures.

Generally, the critical factor of a dragging brake is the adequate dissipation of heat generated during the winching-in or dragging part of the cycle. Previous to the design of the self-energizing handling gear, tests were made by lowering a known load against the brake. It was found that the brake drum could absorb approximately 7½ hp with a leveling-off temperature of about 600 F. which was, of course, excessive. Lowered coefficient of friction, (or fading), and rapid lining wear were pronounced.

Fundamental brake calculations made previous to the construction of the self-energizing handling gear showed that the tangential load imposed on the brake wheel by the spring when the brake was dragging was approximately 350 pounds. This resulted in under four horsepower to be dissipated. Considering, too, that in actual operation the "winching-in" cycle at a maximum is well under 50 per cent of the working day, it was felt that construction of a pilot model with the self-energizing brake was justified. Tests made after construction proved the calculations and the worth of the new handling gear.

Operator Effort Effective

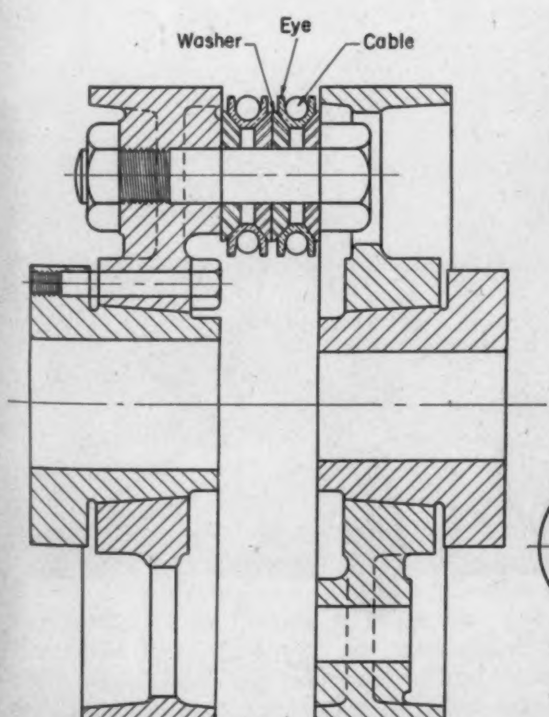
The arrangement permits the operator to utilize his full effort to apply the clutch during the spooling-in cycle. Lever travel is reduced by half since, during this phase of the operation, the operating lever pivots about a fixed point instead of a movable point (brake spring pivot) as in the previous handling gear.

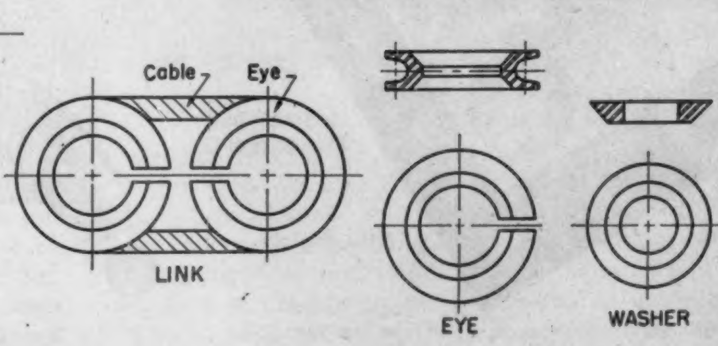
For spooling-out line the operator has only the low-rate brake spring to compress. Since the effort is small, the mechanical advantage was reduced so that the necessarily greater loose end movement required to free spool the two brake bands was obtained. The net result is a handling gear system easy to operate, without excessive lever travel and without excessive generation of heat.

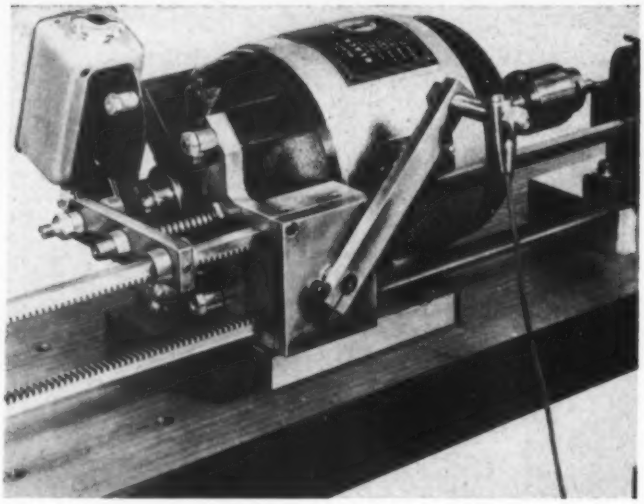
SCANNING the Field For Ideas

Endless cable links, utilized in a novel manner, form the flexible member of the coupling, right, designed primarily for transmitting high torques between connected shafts having as much as five-degree angular misalignment. In the 7-inch diameter unit illustrated, which was tested under three-horsepower pulsating load at 85 rpm, no measurable wear was encountered after a 700-hour run.

Major features of the link design are its simplicity of adjustment and ability to absorb shock and torsional vibration. Should wear occur in the linkage system, tightening of the bolts re-establishes the desired link tension by expanding the link eyes shown in the drawing below. Each flange of the coupling has four bolts alternately spaced with respect to the flanges. On each bolt are two split brass eyes with chamfered bores and four tapered washers so mounted that tightening the bolts increases the diameter of one or both of the eyes. In this way backlash takeup adjustment divides the load between the links, the eye for the looser link expanding to equalize the tension. Invented by H. L. Allen, consulting mechanical engineer, the coupling also has been built with links wound with thin-gage stainless steel strip. The strip was found to be as efficient as the steel cable and less costly inasmuch as the cable must be hand woven.



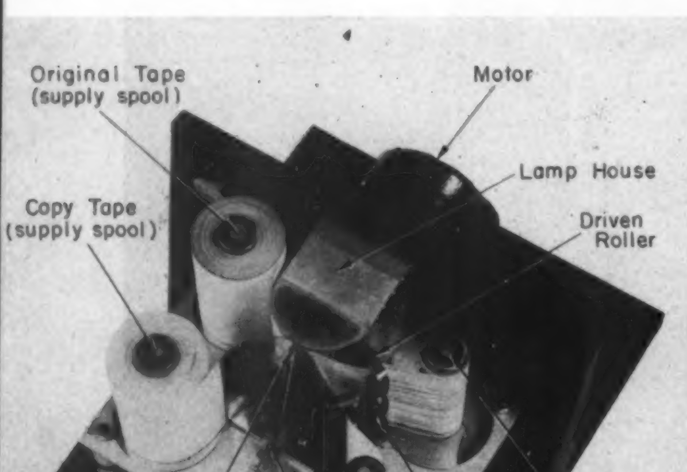


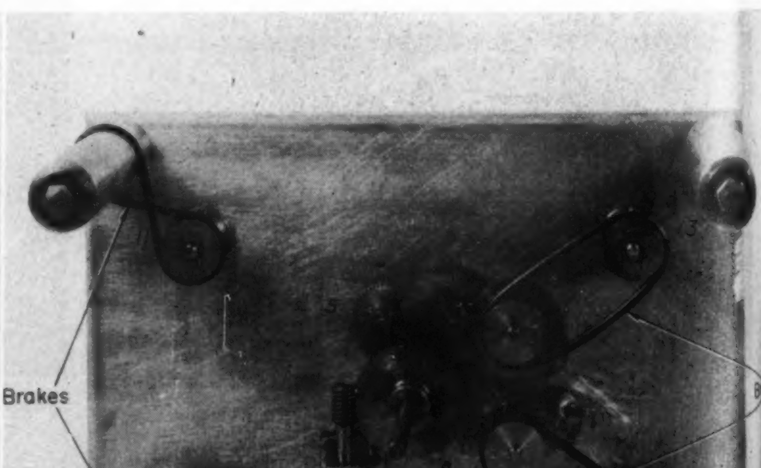


Floating rotor simplifies precision feeding of the spindle for the electric-motor drilling unit illustrated above. Movement of the upper rack by the crank provides the feed through a micrometer adjustment on an arm attached to the rack. With the unit and arm adjusted with respect to the work fixture, precise depth of feed is obtained by adjusting the micrometer screw which engages a thrust bearing on the back end of the spindle. After the feed cycle is completed and the crank released, the spindle returns because of the centering action of the motor magnetic field on the rotor. Designed by Black Drill Co., this traversing shaft motor has a maximum travel of one inch. The bottom rack shown in the illustration feeds a second unit similarly but in the opposite direction for a twin countersinking operation.

Time-base reducer, below, foreshortens or reduces the time scale 100 times to facilitate checking or initial interpretation of test data such as oscillographic records. For instance, recording of certain rapidly varying phenomena requires a chart speed of 2 inches per second, requiring nearly 100 feet to obtain a record of a single run. Plotting a complete detailed analysis of such a record requires approximately two days. Slow variation of any test condition might not become apparent until the analysis is completed. The time-base reducer, however, produces a record that shows quickly the overall picture of the degree of success of the test run, indicating the advisability of proceeding with the current operations or the nature of the changes that should be made for the next run.

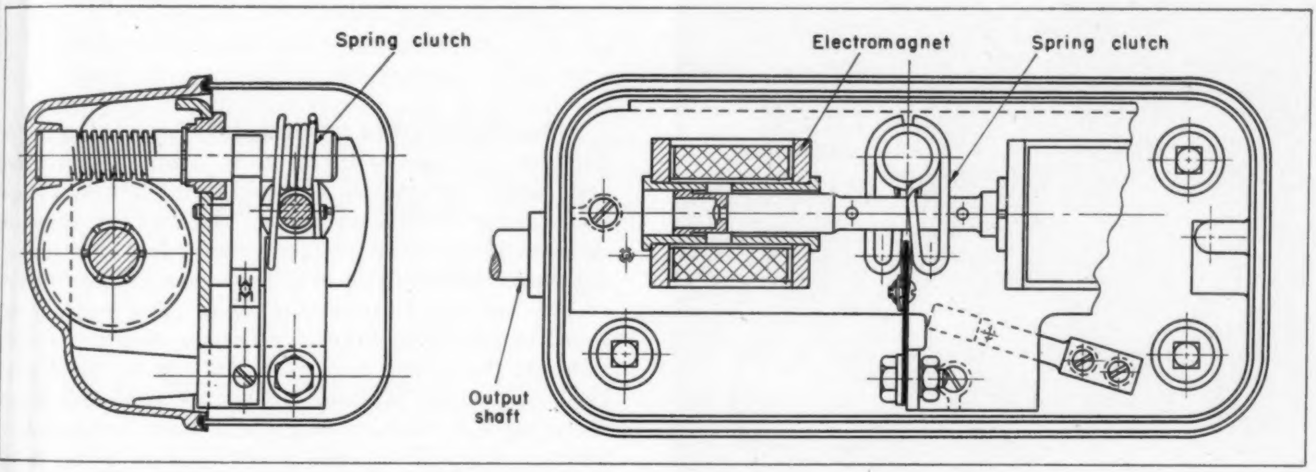
This reducer is a photographic continuous contact printer in which the record and "copy" paper or film are moved at differential speeds past a light slit. Both are drawn by roller feed between two semicylindrical drums. One drum contains a 1/64-inch light slit and the other is pressure loaded by springs to maintain contact of the paper during printing. The light source behind the slit consists of three pilot type lamps arranged in line with the slit. Width of

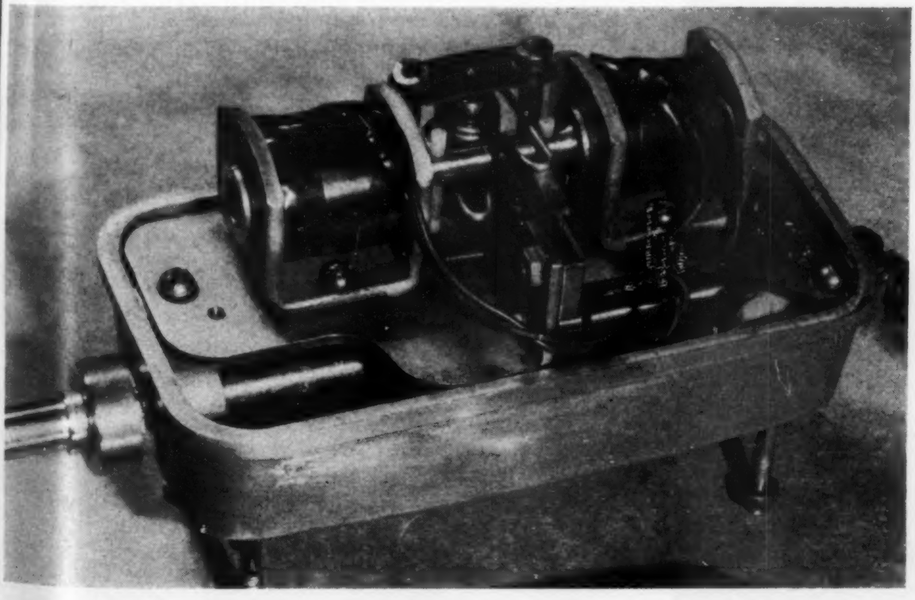




the slit is about the average width of a line trace on the record. As the line crosses the slit it throws a shadow on the paper being exposed, retracing photographically a line which moves transversely with the same magnitude as the original line but with increased slope due to the speed of the printing paper being less than that of the record. Traces having slopes greater than 45 degrees do not reproduce clear traces, vertical traces not being resolvable because of approximate coincidence with the slit itself.

Power for the reducer is supplied by a 1/20-horsepower motor driving two sets of gears through a worm-gear reducer. Each gear train drives a feed roller, the chart feed roller being driven ten times faster than the printing roller. When the copy print is rerun to make a positive print, the latter also copies at a 10:1 ratio, making a final reduction of 100:1 of the original record. The two take-up spools are connected by pulleys and rubber belts to their respective roller feeds. These belts are fitted to provide take-up tension only. The two supply spools are similarly fitted with pulleys and belts, friction braked to stationary pulleys. This drive arrangement is shown in the bottom view of the unit.





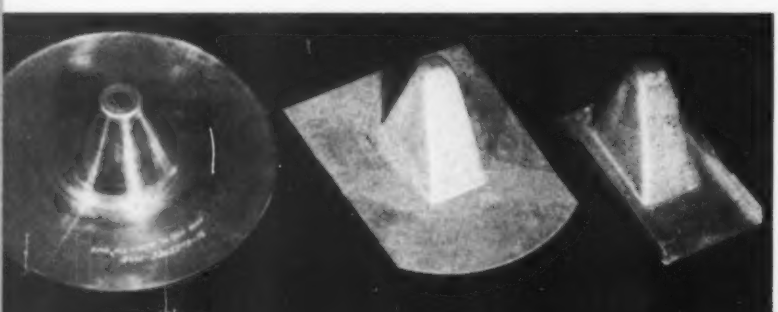
Two-way spring clutch and magnets motivate the unusual reversible motor above. It starts instantly upon closing the contacts and stops similarly without a coasting period, making it adaptable to positioning applications. Inasmuch as the magnet impulses are imparted at the frequency of the a-c supply, seemingly continuous operation of the slow-speed output shaft is obtained from the impulses.

When energized by one of the magnets, the spring clutch, shown in the drawing above, turns the wormshaft by gripping the shaft with each minute pulse in one direction and slipping in the other direction. The second magnet produces reverse operation by engaging the opposite end of the clutch spring. Having no backlash, the clutch is free in one direction but grips rigidly in the other during actuation of the shaft in either direction.

High starting torque is attained because the rotation of the shaft is independent of the speed of the magnet armature until equilibrium is reached between input and output. In this mechanism engineered by the Viking Tool & Machine Corp., the motor may be controlled by several methods to vary its speed. One method involves inserting a half-wave rectifier in the circuit which approximately doubles the speed and the other is to insert a variac or rheostat in the circuit. With a 50:1 reduction gear the motor runs on 60-cycle current at about ¾ rpm.

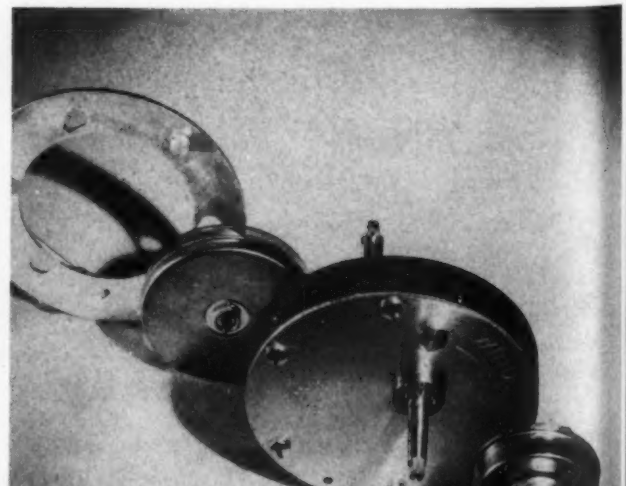
Ideas

Developed by H. W. Snook at Douglas Aircraft Co., the prespinning method obviates six sets of staging dies for the switch box illustrated. With the blank prespun into a neutral hat shape, only three sets of dies are required to finish the part. Distribution of the metal in the spinning is so controlled that rupture or breakage in final forming is rare. Form blocks for spinning are usually birch, maple, masonite, aluminum, Kirksite, or steel, depending on the severity of the spinning and the number of parts desired. Alloys suitable for spinning include aluminum, brass, terne plate, 1025 carbon steel, and 18-8 stainless, some requiring annealing due to workhardening. There is no set formula for the prespun shape but experience is narrowing the number of trial and error prespinning that was necessary when the process was inaugurated.



Spun neutral shapes, above, eliminating several sets of multiple staging dies, require only finish forming and trimming operations. Particularly advantageous for short production runs for parts such as the switch box, above, the spinning operation may be utilized to control section thickness, minimizing local thinning and possible rupture during final forming.

Interchangeable assemblies, for which gear reductions are built as turret units, simplify production and reduce inventories for the spring-driven escapement motor illustrated below. Two motor types, 8-day and 45-day wind, fit a universal adapter or mounting ring. Either type may drive one of nine turrets to give ten speed ranges for each. The 8-day motor speed ranges from 2 hours to seven days per revolution and the 45-day motor from fifteen to thirty days. Designed by the Macnick Division of Rockwell Manufacturing Co., these power units were developed for driving charts in recording instruments or operating electrical contacts in time-cycle controls. The drive mechanism is hermetically sealed with neoprene gaskets in a two-piece aluminum case and the square drive arbor is stainless steel. Simply and quickly attached, a turret is assembled to an arbor by locking it in place with a quarter turn.



Pneumatic Standards for Industrial Equipment

PRESENTED IN THIS DATA SHEET are the recently approved Pneumatic Standards for Industrial Equipment compiled by the Joint Industry Conference (J.I.C.) of seven co-operating groups of manufacturers. The standards apply to all types of air-actuated industrial equipment, including but not confined to machine tools. Although these standards, like any others, are subject to further modification, the editors of MACHINE DESIGN feel that their publication in full at this time will serve a useful purpose in stimulating thinking, promoting more uniform practices, and leading to improvement in the standards themselves

General

A0.1—*a.* The purpose of these Standards is to provide specifications for the application of pneumatics to industrial equipment which will promote safety of personnel, uninterrupted production, long life of the machine, equipment or tool, and will not limit or inhibit advancement in the art of pneumatics.

b. Where new developments in pneumatics will perform in a manner equal to these specifications they will be considered as equivalent.

A0.2—These Standards are proposed to cover pneumatic equipment in general applications. Any exceptions to these Standards will be as agreed upon between supplier and purchaser.

A0.3—The purchaser shall specify such additional details as are needed to meet unusual operating conditions.

A0.4—*a.* The builder shall submit necessary performance specifications on pneumatic equipment, including symbolic or schematic diagrams in accordance with paragraph A1.1.4 *a* for purchasers' approval. Final diagrams and text shall conform to the pneumatic equipment shipped.

b. Part numbers shall be provided for all types of pneumatic equipment. Seal manufacturer's catalog number shall also be provided where sealing devices are commercially available.

A0.5—All pneumatic equipment shall be installed and used in accordance with the recommendations of the manufacturers thereof.

A0.8—All industrial equipment at the completion of assembly and before shipment shall be given complete performance tests to determine conformity with equipment specifications. There shall be no evidence of external leakage.

A0.9—A pressure-time diagram taken under operating conditions including data concerning maximum rate of pressure rise shall be furnished when specified on purchase order.

A0.10—Pneumatic equipment shall be of proved engineering design, fabricated of material of accepted quality and with workmanlike finish; and must be capable of performing efficient, reliable service in the installation for which it is provided.

A0.11—Pneumatic equipment and piping shall be accessible and mounted in a position that will not interfere with machine adjustments or maintenance of electrical, hydraulic, or mechanical equipment. (See A2.6.3; A4.2.6 and A6.3.4).

Diagrams

Circuit Diagrams

A1.1.1—*a.* SYMBOLIC DIAGRAM: A drawing or drawings showing by means of approved standard symbols each and every piece of pneumatic apparatus, including all interconnecting lines.

b. SCHEMATIC DIAGRAM: A drawing or drawings showing the functional construction of all valves, controls, and actuating mechanisms including all interconnecting lines.

A1.1.2—SYMBOLIC DIAGRAMS:

a. Symbolic diagrams shall be furnished with industrial equipment. (Schematic diagram acceptable.)

b. A copy of the diagram shall be firmly attached to the inside cover of the pneumatic control enclosure or pneumatic diagram enclosure, either by adhesive material or by permanent data pocket or clip.

A1.1.3—SCHEMATIC DIAGRAMS:

a. When requested on purchase order, a schematic diagram shall be supplied.

b. When requested on purchase order, a schematic diagram shall show and describe the flow paths for each phase of each complete cycle.

c. When requested on purchase order, a schematic diagram shall show in color only those portions of the circuit which are functioning at each single phase of the cycle. (See A1.4)

A1.1.4—ALL DIAGRAMS:

a. The descriptive text shall explain the function of all components of the circuit and the sequence of the operations. Either the diagram or descriptive text shall include in addition to the sequence of operations, the following when applicable:

1. All pneumatic equipment shall be identified. When possible, catalog number and manufacturer's name shall be shown
2. Size of piping
3. Diameters of pistons and rods, length of stroke, and

The Joint Industry Conference consists of the following co-operating groups: Hydraulic Equipment Manufacturers; Industrial Equipment Users; National Machine Tool Builders' Association; Packing and Seal Manufacturers; Pneumatic Equipment Manufacturers; Press Manufacturers; and Tubing and Fitting Manufacturers.

Glossary of terms used in pneumatic specifications

Actuator: A device to convert fluid energy into mechanical motion.

Air: Air as referred to in these standards means the pressure medium for pneumatic applications.

Automatic Controls: Those actuated in response to the cycle of the equipment.

Back Connected: Where piping connections are on normally unexposed surfaces of pneumatic components.

Channel: An air passage the length of which is large with respect to its cross-section dimension.

Circuit: An arrangement of component parts or pneumatic equipment interconnected to a specific appliance or appliances.

Cleaner: A device for the removal of solids from a fluid wherein the resistance to motion of such solids is a straight line.

Compartment: A space within the base, frame or column of the equipment.

Compressor: A device for converting mechanical energy into air energy.

FIXED DISPLACEMENT: A compressor which delivers a relatively constant volume of air per cycle.

VARIABLE DISPLACEMENT: A compressor in which volume per cycle can be varied.

Cylinder: A linear motion device in which the thrust or force is proportional to the effective cross sectional area and to the pressure differential.

Cylinder, Single Acting: A cylinder in which the air pressure can be applied in only one direction.

PLUNGER TYPE: A cylinder in which the internal element is of a single diameter, and upon which the seal applied is of the contracting type.

PISTON TYPE: A cylinder in which the internal element is one or more diameters, and the seal is of the expanding type.

Cylinder, Double Acting: A cylinder in which air pressure can be applied in either direction.

SINGLE END ROD: A cylinder with a rod extending from one end.

DOUBLE END ROD: A cylinder with two rods, one extending from each end.

Enclosure: A housing for pneumatic apparatus.

Filter: A device for the removal of solids from a pneumatic system wherein the resistance to motion of such solids is in a tortuous path.

Fluid: Any material that will take the shape of the container in which it is placed.

Front Connected: Where piping connections are normally on exposed surfaces of pneumatic components.

Pneumatic Panel:

1. PNEUMATIC MOUNTING PANEL: A plate on which may be mounted a number of pneumatic actuating components.
2. PNEUMATIC CONTROL PANEL: A grouping of pneumatic control units mounted to form one assembly on a plate, or in a casting, and having a single mounting surface.

Line: A tube, pipe or hose which acts as a conductor of air.

Line, Exhaust: A return line which carries power or control actuating air back to the atmosphere.

Line, Joining: A line which crosses or connects with another line on a diagram and which also connects in construction. The junction of the lines shall be indicated by a dot which must be at least three times the width of the line.

Line, Passing: A line which crosses another line on a diagram, but which does not connect in construction. Line on a diagram to have no swell at crossing.

Line, Pilot: A line which acts as a conductor of control actuating air.

Line, Working: A line which acts as a conductor of power actuating air.

Lubricator: A device for adding lubricant to the actuating medium.

Manual Controls: Those actuated by the operator, regardless of means.

Mass Production: For a model run, or an indefinite period of time.

Motors and Cylinders: Devices for converting air energy into mechanical energy.

Motor, Oscillating: A motor producing a maximum angular rotative movement of less than 360 degrees in either direction.

Motor, Rotary: A motor producing rotary motion having torque output proportional to the displacement per revolution and to the pressure drop between intake and exhaust ports.

FIXED DISPLACEMENT: A rotary motor in which the displacement per revolution is not adjustable.

VARIABLE DISPLACEMENT: A rotary motor in which the displacement per revolution is adjustable.

Muffler: A device for reducing exhaust noises.

Passage: A machined or cored connection which lies within or passes through a pneumatic component, and which acts as a conductor of air.

Phase of Cycle:

0. NEUTRAL
1. RAPID ADVANCE: The approach of the tools or workpiece to the feed position.
2. FEED OR PRESSURE STROKE: The portion of the cycle where work is performed on the workpiece.
3. DWELL: The portion of the cycle where the feed rates or pressure stroke is stopped.
4. RAPID RETURN: The return of the tools or workpiece to the cycle starting position.

Port: An opening at a surface of component; e.g., terminus of a passage. May be internal or external.

Port, Valve: A controllable opening between passages; e.g., one which can be closed, opened or modulated.

Positive Position Stop: A structural member which definitely limits the working motion at a desired position.

Positive Safety Stop: A fixed structural member which confines maximum travel within the design limits of the machine or equipment.

Pressure-Time Diagram: A graphical presentation of pressure in pounds per square inch plotted against time for a complete cycle of equipment.

Restriction: A device which produces a deliberate pressure drop or resistance in a line or passage by means of a reduced cross-sectional area.

CHOKE: A restriction, the length of which is relatively large with respect to its cross-section dimensions.

ORIFICE: A restriction, the length of which is relatively small with respect to its cross-section dimensions.

Schematic Diagram: A drawing or drawings showing the functional construction of all valves, controls, and actuating mechanisms.

Sealing Device: A part or assembly of parts used to prevent leakage between two or more other parts.

Separator: A device to separate water or other materials of different specific gravities from the actuating medium.

Subplate (Back Plate): An auxiliary mounting for a pneumatic actuating component to which piping connections are mounted.

Surge: A transient rise in air pressure in the circuit.

Symbolic Diagram: A drawing or drawings showing by means of approved standard symbols each and every piece of pneumatic apparatus including all interconnecting lines.

Trip Device: A mechanical element for the actuation of a position control.

Unit Production: One piece per setup.

estimated required force of cylinders when other than maximum pressure is applied

4. Time of cycle when pertinent, e.g., time range of cycle exclusive of loading

5. Operating pressures

6. Horsepower and revolutions per minute of each pneumatic drive

7. Delivery and consumption in cubic feet of free air per minute

8. Storage or surge tank capacity in cubic feet

9. Displacement, speed range, and torque rating of each pneumatic motor

10. Data or text, or both, shall show operations performed and characteristics of any actuating electrical equipment used.

b. The related text and the diagram shall be shown on the same or facing pages.

c. All diagrams shall be marked with serial number or with the purchaser's order number of that equipment, or, if none, with some other symbol which will identify the particular apparatus to which the diagram applies.

A1.1.5—Requested number of copies of final diagrams and texts shall be forwarded by mail or in person to an individual delegated by the purchaser not later than the date on which the shipment is made.

Standard Symbols

INTRODUCTION: *a.* Standard symbols shall be used for symbolic diagrams and for showing flow paths in schematic diagrams.

b. Direction Control Valves: The flow condition which is shown closest to the operating device is that which exists when the device is operated.

c. The standard symbols and sample drawing of a pneumatic circuit utilizing the standard symbols to indicate the nature and function of component parts in the circuit are shown on the following pages.

Legend Code

A1.4.1—COLOR CODE:

Intensified pressure	Black
Supply pressure	Red
Charging pressure	Intermittent red
Reduced pressure	Intermittent red
Metered flow	Yellow
Exhaust	Blue
Intake	Green
Inactive	Blank

A1.4.2-PATTERN CODE

Function	Pattern No.	Suggested zip-a-tone or equivalent pattern
Intensified Pressure	7R	
Supply Pressure	177	
Charging Pressure Reduced Pressure	52-0	
Metered Flow	63	
Exhaust	75	
Intake	16	
Inactive	Blank	

A1.4.3—DEFINITION OF FUNCTIONS:

Intensified pressure	Pressure in excess of supply pressure and which is increased by a booster or intensifier
Supply pressure	Power actuating air
Charging pressure	Compressor - inlet pressure that is higher than atmospheric pressure
Reduced pressure	Auxiliary pressure which is lower than supply pressure
Metered flow	Controlled flow rate
Exhaust	Return of power actuating medium to atmosphere
Intake	Subatmospheric pressure, usually on intake side of compressor
Inactive	Air pressure which is within the circuit but which does not serve a functional purpose during the phase being represented.

Diagram Size

A1.5.1—All diagrams shall be of such size as to clearly disclose all components of the systems. Diagrams shall be 8½ x 11 in. or folded to that size.

Pneumatic Controls

Definition of Controls

a. Manual controls are those actuated by the operator, regardless of means.

b. Automatic controls are those actuated in response to the cycle of equipment.

Protection

A2.3.1—Over-pressure and under-pressure regulation shall be provided on pneumatic circuits where required. The adjustments on the pressure control units shall be accessible. Provision shall be made to permit locking enclosure or compartment in which pressure control unit is mounted, or protective locking type adjustments shall be provided to prevent tampering.

A2.3.2—All pressure controls shall be so constructed as not to be adjustable outside nominal working ranges, and shall be marked to indicate nominal minimum and maximum pressures.

A2.3.3—Where there is more than one source of supply on the equipment and possible damage may be caused by the stoppage of any one source, interlocks shall be provided so that the stoppage of one source in the circuit will not endanger or damage the equipment.

A2.3.6—Where loss of supply pressure may result in damage to equipment, loss of accuracy, or danger of injury to personnel, means shall be provided to prevent operation under those conditions.

Controls, Circuit

A2.4.1—When shifting from rapid traverse to feed, the starting position of the feed rate shall be maintained consistent with the type of service required.

A2.4.2—Pneumatic circuits shall be so designed as to minimize air consumption, the generation of heat and to conserve power.

A2.4.4—In each electrically controlled circuit, in case of solenoid failure, means shall be provided to prevent damage to equipment, loss of accuracy, and danger of injury to personnel.

A2.4.5—Neutral valve positioning controls shall be independent of and unaffected by other controls.

*A2.4.6—Where practicable, feed and cycle limiting trip devices on mass production equipment shall be so arranged that the feed distance can be set and sealed. After feed distance is preset, the control mechanism shall be adjustable as a unit.

A2.4.7—*a.* Positive position stops shall be so designed that the dwell time will not be affected when positive position stops are reset.

b. Positive safety stops for machine slides shall not be adjustable, but have provision for replacement.

A2.4.8—On industrial equipment operated at elevated temperatures or with toxic materials, the circuits shall be so designed and located that they can be serviced while equipment is operating under these conditions.

A2.4.9—Pneumatic circuits shall be so designed, constructed and applied that unnecessary surge pressures are minimized at their source, and equipment shall be of adequate strength to withstand all operating pressures and surge pressures.

A2.4.10—Pneumatic circuits shall be so designed as to prevent uncontrolled movement of the equipment-op-

* Standard specifications noted with an asterisk were written as a guide in indicating the type of engineering that is considered to be desirable in new developments and in the re-engineering of equipment.

LINES

Line, working	Direction of flow	Plug or plugged connection — ×
Line, pilot	Line, passing	Testing station (gage connection) — T S
Exhaust	Line, joining (tee, cross etc. dot to be 3 x W)	Power take-off — P O
Line, flexible	Storage or surge tank	Restriction, fixed
Connector (dot to be 3x width of associated line)		

COMPRESSORS

Compressor, single, fixed displacement — CF	Compressor, single, variable displacement — CV

MOTORS AND CYLINDERS

Motor, rotary, fixed displacement — MF	Cylinder, single acting plunger type
Motor, rotary, variable displacement — MV	Piston type
Motor, oscillating — MO	Cylinder, double acting single end rod
Cylinder, rotating, air — R	Double end rod

MISCELLANEOUS UNITS

Motor, drive, electric — ELEC MTR	Intensifier — INT	Filter — FILT	Pressure switch — PR SW
Lubricator — LUB	Accumulator — ACC	Separator — SEP	Pressure gage
Spring	Shaft, rotating	Component enclosure	

VALVES

Valve, check	Valve, basic symbol (insert model no. for special valves)
Valve, restriction, variable	Method of indicating internal flow

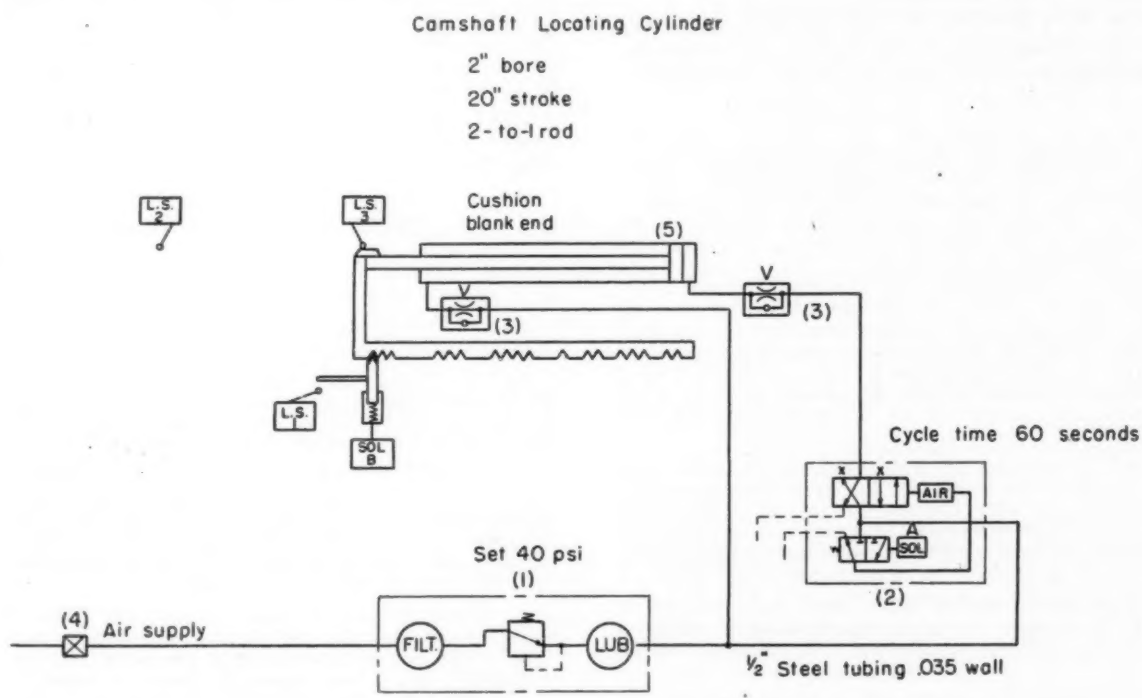
VALVE EXAMPLES

Valve, manual shut off	Valve, flow control pressure compensated — V (PC)
Valve, maximum pressure	Valve, shut off 2 position - 2 connection
Valve, relief remotely operated	Valve, directional 2 position - 3 connection
Valve sequence directly operated	Valve, directional 2 position - 4 connection
Valve pressure regulator	Valve, directional 3 position - 4 connection Closed center

METHODS OF OPERATION

Control, basic symbol	(blank box)	Control, manual	MAN	Control, solenoid	SOL
Control, centrifugal	CENT	Control, mechanical	MECH	Control, solenoid hyd. pilot operated	SOL HYD
Control, compensator	COMP	Control, motor electric	ELEC MTR	Control, solenoid air pilot operated	SOL AIR
Control, compensator pressure	PR COMP	Control, motor air	AIR MTR	Control, thermal	THERM
Control, compensator temperature	TEMP COMP	Control, pilot hydraulic	HYD	Control, pilot hyd. differential area	HYD DIFF
Control, cylinder	CYL	Control, pilot air	AIR		
Control, detent	DET	Control, servo	SERVO		

SAMPLE PNEUMATIC CIRCUIT



(1) Operator loads and clamps part manually and pushes cycle start buttons

(2) Solenoids "A" and "B" are energized and when L.S.-1 is tripped solenoid "B" is de-energized

(3) Cylinder moves forward until wedge encounters next notch in rack

(4) Cylinder dwells at this point until electric timer times out and solenoid "B" is again energized and fixture moves to next notch in rack

(5) This cycle is repeated until L.S.-2 is tripped

(6) L.S.-2 energizes timer 2, when timer 2 times out solenoid "A" is de—energized and solenoid "B" is energized and cylinder returns tripping L.S.-3 de-energizing solenoid "B" completing cycle

erating devices in all phases of the equipment cycle including the idling phase.

A2.4.11—Pneumatic circuits shall be so designed, and when necessary incorporate suitable dash-pot devices, as to adequately compensate for load variations and their effect on the compressible actuating medium in relation to the intended service.

A2.4.12—Circuits having two or more heads, slides or actuators shall be provided with means to insure operation of all units in proper relation regardless of load variation.

Enclosures and Compartments

A2.5.1—For the purpose of these Standards, an enclosure means a housing for pneumatic apparatus. A compartment means a space within the base, frame, or column of the equipment.

A2.5.2—Automatic controls that might be damaged by exposure to dirt or dust or stock moving equipment shall be adequately protected and readily accessible.

A2.5.4—Doors and other parts of compartments for automatic controls shall be sheet steel not less than 14 USS gage, or cast metal not less than ¼-in. thick. Compartments for automatic controls shall be completely isolated from coolant, cutting or lubricating oil, chips, and dirt.

A2.5.6—*a*. When enclosures for control devices are required they shall have hinged covers which swing horizontally and shall be held closed with screws which require the use of a tool to remove.

b. Means shall be provided for locking when requested.

A2.5.7—The arrangement of enclosures for mounting controls shall permit ample room for proper maintenance.

Mounting

A2.6.1—*a*. ALL CONTROLS: All controls shall be mounted so as to minimize the possibility of damage from floor trucks or other stock moving equipment. They shall be protected against falling objects and operator's accidental movements. No controls shall be placed in high heat areas or in locations subject to any harmful conditions unless properly protected.

b. MANUAL CONTROLS: All manual controls shall be mounted within reach of the operator when in his normal operating position. Operator shall not be required to reach past revolving spindles or moving tools to reach manual controls. (See Paragraph A7.1.2)

c. AUTOMATIC CONTROLS: Automatic controls shall be so located and protected as to prevent inadvertent operation.

A2.6.2—All pneumatic controls shall be mounted two feet or more above the bottom of the equipment providing such mounting will not impair the proper functioning of the circuit. (This paragraph is written as a guide for all new machinery and equipment with the understanding that any deviation will be in accordance with A0.2).

A2.6.3—Pneumatic control units shall be mounted in positions which provide complete accessibility to all units for maintenance without interfering with adjacent equipment.

A2.6.5—All industrial equipment shall incorporate an emergency stop control readily accessible from operator's normal position.

A2.6.7—Piping shall not be used as a means for supporting valves or equipment where vibration may prove detrimental to piping.

A2.6.8—Automatic controls shall be mounted as close to actuator as practicable to keep working lines as short as possible to prevent waste of air and excessive actuating time lag. Mounting should be such that adherence to the above does not violate Paragraph A2.6.3.

Compressors

Mounting Compressors

A3.3.1—Compressors or subassemblies including the compressors, shall be readily accessible for maintenance.

A3.3.2—Compressors shall be mounted where they are adequately protected from damage.

A3.3.3—When compressors and other apparatus are mounted inside a compartment or enclosure, sufficient space must be provided to assure complete accessibility to all units.

A3.3.4—Compressor compartments must be clean, free of moisture and adequately vented. All openings must be of sufficient height above the floor to be protected to such an extent that dirt, chips, coolant, etc., cannot enter, especially when floor is swept or washed.

A3.3.5—Direct-coupled compressors must be properly aligned and securely mounted in such a manner as to assure alignment under normal operating conditions. When mounted inside column or base of equipment, foot-mounted compressors and driving motors shall be mounted on a subbase of adequate strength which can be easily removed for servicing. The coupling must have adequate capacity to transmit the power required.

Compressor Nameplates

A3.5.1—The following information shall be permanently indicated on each compressor:

a. Manufacturer's name and address
b. Manufacturer's identifying part number
c. Manufacturer's serial number if any.

A data sheet or catalog providing complete information on speed limits, pressure range, delivery and horsepower required shall be provided with the specifications.

A3.5.2—Whenever a compressor is mounted so that its nameplate is not readily visible, a plate with duplicate information shall be provided where it can be readily seen. Such duplicate plate shall be marked "duplicate." The nameplate shall not be removed from the compressor.

A3.5.3—The direction of rotation of each compressor shall be clearly indicated on the compressor where it can be readily seen.

A3.5.4—All necessary valves for removal of any compressor while other compressors are still in operation shall be provided.

Piping, Fittings, and Passages

(Piping includes all pipe, tubing, hose and fittings. Passages include all air conductors other than piping.)

A4.1.1—When requested on purchase order, a piping layout or photograph shall be furnished.

A4.1.3—Piping shall have adequate strength to withstand additional pressure, including surge pressure, within the safety limits of the entire circuit. A factor of safety of at least eight (8) over normal working pressure is recommended except for flexible hose. (A2.2.4).

A4.1.5—Piping leads in working circuits between actuating devices and feed control devices shall be so constructed as to rigidly confine a minimum volume of air, the purpose being to maintain constant, controlled motion and minimize the effect of varying forces.

A4.1.6—*a*. Solderless pipe connections shall be used, e.g., flared, flareless, self-flaring, flanged, or equivalent.

b. When flared type fittings are used (which require a flaring tool), the angle of flare of tubing shall be 37 degrees from the centerline, 74 degrees included angle, and flare dimensions in accord Army-Navy Design 10061.

**c*. All pipe threads shall be Dryseal American (National) standard taper pipe threads (N.P.T.F.).

d. Fittings with straight threads, which incorporate seals that seal with pressure, may be used in place of pipe thread fittings.

e. All piping connections shall be so designed and installed as to permit quick removal and reassembly by means of hand tools.

f. Tubing outside diameter sizes shall increase in 1/16-in. increments from ⅛-in. to and including ⅜-in., and in increments of ⅛-in. above ⅜-in. diameter up to and including 1-in. diameter and in ¼-in. increments above 1-in. diameter.

g. Fittings with restricted or stepped-up passages are not recommended.

A4.1.7—All piping, piping fittings, passages, storage or surge tanks, cored holes or drilled holes must be free of burrs, or foreign matter which might cause damage to any pneumatic unit. Sharp edges shall be removed whenever they adversely affect the flow of air.

A4.1.8—Whenever practicable, each piping run must be integral and continuous from one piece of apparatus to another. Piping runs must be removable without dismantling equipment components and without bending tubing or spring it in a manner to damage it. All rigid piping must be securely supported at frequent intervals to avoid vibration or movement. Points of contact or support on the outside of the piping shall be so designed that they will not result in damage to the piping. Piping shall not be welded to supports.

A4.1.9—When high or extra high pressure piping is used all connections shall be welded to steel flanges; or connections may be used which are equal in performance and ease of assembly. Flanged connections shall use sealing devices that seal with pressure. (All welded joints shall be stress relieved.)

A4.2.4—For all portions of piping circuits subject to movements in which flexible hose is required, oil-resistant hose shall be used and shall be fabricated to a pressure safety factor of at least five (5) with respect to the circuit in which it is used. Flexible hose shall have vertical conections and shall have sufficient slack to avoid sharp flexing and straining. Horizontal connections are acceptable only if hose is adequately suported.

A4.2.6—Piping shall not be placed where it will interfere with the adjustment, repair or replacement of control units, and piping connections shall be readily accessible for maintenance.

A4.2.7—When construction of equipment necessitates shipment or moving in sections, thorough precautions shall be taken to adequately protect piping runs, whether they be left in place on one of the equipment sections or whether they are removed and separately boxed for shipment. All male threads shall be sleeved, and all piping openings or female threads have suitable closures.

A4.2.8—Piping shall be so arranged that pressures may be tested at accessible locations, preferably at built-in checking stations which shall be indicated on diagrams.

Identification of Piping and Checking Stations

A4.4.1—Piping runs and checking stations shall be permanently identified to correspond with diagram.

Valves, Accessories, and Devices

Filters and Lubricators

A6.1.1—Means shall be provided for the removal of deleterious materials from the pneumatic system which would be detrimental to the operation of the equipment.

A6.1.2—Suitable means of air line lubrication shall be provided when practicable.

A6.1.4—Filters shall be of such construction that the filter medium can be removed for cleaning or replacement without disturbing the piping. When practicable, suitable means shall be provided to indicate when filter should be cleaned.

A6.1.5—Intake cleaners shall be provided and shall be of ample size to clean more than double the capacity. Suitable means shall be provided to remove cleaners.

Sealing Devices

A6.2.1—All sealing devices must be of such materials that are not be adversely affected by air or lubricants.

A6.2.3—Sealing devices on reciprocating or rotating shafts shall prevent leakage, except that required for lubrication of such devices under working conditions without damaging shafts.

A6.2.4—Clearances in sealing glands shall be such as to prevent undue extrusion of the sealing material.

A6.2.5—Stuffing boxes for automatic packing shall be so designed as to prevent adjustment beyond the functional limits of the packing.

A6.2.6—Industrial equipment shall be so designed as to facilitate easy replacement of packings.

A6.2.7—J.I.C. Standard part numbers shall be provided for all packings. If none are available, principal dimensions such as inside diameter, outside diameter, stack weight and material shall be provided.

Cylinders and Pistons

A6.3.1—Cylinder bores having fitted pistons shall be finished in a manner consistent with the type of service intended and shall be free of porosity or other defects.

A6.3.2—Cylinders shall be located in alignment with work slides, and shall be such that no side or radial load shall occur on piston rod or ram, unless other suitable provisions are made to take such loads.

A6.3.3—Ends of cylinders shall utilize sealing devices that do not leak under manufacturers' intended operating conditions. Cylinder external sealing devices shall be readily accessible for servicing.

A6.3.4—*a*. Cylinders shall be accessible for servicing. **b*. Cylinders shall be separate, not cast integral with equipment.

A6.3.6—Piston rods shall be adequately protected by installation of cleaning and sealing devices. Material and hardness of piston rod shall be such as to prevent scoring.

A6.3.7—When pistons are assembled to rod they shall be positively locked.

A6.3.8—When necessary, cushions shall be provided on cylinder ends.

Valves

A6.4.1—Adjustable valves shall be arranged so that wires and seals can be used after final adjustments and locking.

A6.4.3—When practicable subplate mounted valves (or equivalent) shall be used so that their repair or replacement can be made without disconnecting individual air lines. Seals of the pressure sealing type or seals of a type that will provide the equivalent of pressure medium sealing type are recommended. Means shall be provided so that subplate mounted valves cannot be improperly assembled to the subplate.

A6.4.5—The operation of a valve shall not contribute to detrimental surges in the pneumatic system.

A6.4.6—On vertical or inclined slides or rams means shall be provided to prevent rapid drop at air shutoff.

A6.4.7—Solenoid operated valves shall be so designed, constructed and installed as to eliminate hammering.

Storage or Surge Tanks

A6.5.1—Tanks shall be so constructed as to conform to State regulations. Where no regulations are in effect, they shall be so constructed as to withstand at least five (5) times the operating pressure of the system.

A6.5.2—Tanks shall be so designed and constructed that they cannot be disassembled while containing an unsafe charge. Means shall be provided for safely relieving air pressure.

A6.5.3—Pneumatic circuits incorporating tanks shall be so arranged that the system can be bled at the low point of the circuit.

A6.5.4—See Paragraph A7.1.3 of the "Safety" section.

A6.5.5—Where applicable, all tanks shall incorporate suitable automatic condensate dump valves and suitably protected water indicators.

Storage or Surge Tank Nameplate

A6.6.1—The following information shall be permanently indicated on each tank:

a. National board serial number
b. Name of manufacturer
c. Maximum allowable working pressure
d. Manufacturer's serial number
e. Year built.

Safety

A7.1.1—Flexible lines shall be restrained or confined, if their failure might constitute a hazard to personnel.

A7.1.2—Operator shall not be required to reach past revolving spindles, moving tools, or moving machine or equivalent elements to reach manual controls.

A7.1.3—Pneumatic circuits incorporating storage or surge tanks shall be so interlocked as to vent or isolate pressure when power is shut off. On circuit applications where tank pressure is isolated full information shall be given on or near tank for proper servicing.

DESIGN ABSTRACTS

Controls for an Automatic Lathe

By M. S. Curtis
*The Warner and Swasey Co.
Cleveland, O.*

THE single-spindle automatic-chucking machine shown in *Fig.* 1 was designed with the following objectives in view: reduction of tooling costs; ease of set-up, both of tools and of controls; and flexibility, which affects both set-up time and tooling costs. Let us first look at attempts to reduce cost, both of machine and tools. Not much can be done about the former, except that it is axiomatic that the simplest way to drive turret and cross slides on an automatic is by cam. Usually the trouble is that cam feed either reduces flexibility, or by making cam changes necessary, increases costs and set-up time.

We endeavored to do three things: (1) Maintain flexibility without cam change; (2) Make the machine so that tool holders could be so mounted and adjusted as to permit a large variety of work to be done with standard tooling; (3) Make the machine rigid and accurate so that costly special tooling measures would not be required to overcome deficiencies in the machine itself.

Various methods were considered for mounting and locking the main tool slide or pentagon turret, *Fig.* 1. The aim was an arrangement that would be rigid, minimize the effect of heat, have accurate locating means, and have longitudinal adjustment of tools to eliminate tool overhang.

On the machine, this turret is integral with a bar extending across the column of the machine directly above the spindle. There are longitudinal dovetail slots in each turret face which permit easy application of the turret tools. It is only necessary to slide the turret tool holders into these dovetail slots and tighten up a gib. This design permits endwise adjustment of the tool holders, and allows the use of standard tool holders for a majority of jobs. The turret is cam driven, the cam being mounted within the column of the machine. The cam is driven forward and reverse for a like movement of the turret rather than unidirectional as on most single spindle automatics. Its auto idle motions are controlled by a separate reversible motor, so that in setting up the machine, the operator has pushbutton control of forward and reverse rapid traverse. Therefore no hand crank is necessary to back tools away from the work, if that should become necessary before the turret has advanced to the end of its stroke. The reversing motor for traverse movements and its

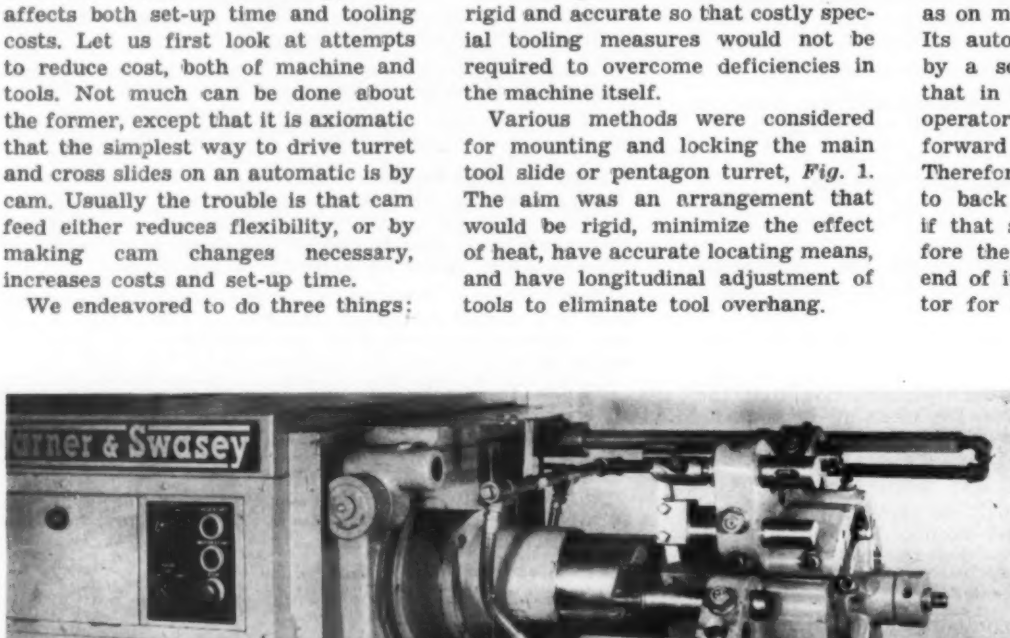


Fig. 1—Single-spindle automatic - chucking automatic lathe. Actions of turret and cross - slide tools are electrically controlled through switch arrangements such as shown in *Figs.* 2 and 3

pushbutton control give the desired flexibility to turret travel in setting up the machine.

A spider with an arm for each turret face is fastened to the turret bar inside the column and traverses between a pair of hardened steel blocks while the turret is being operated. The bar bearings and the spider bearings are the only ones used to locate and guide the turret, whereas the conventional type turret has at least four bearings—one each for longitudinal guidance, traverse guidance, vertical location, and index location. To index the turret it is only necessary to pull the spider out from between the hardened steel blocks.

Since chuck, turret and cross slides are all at one end of the machine bed, they are readily accessible from the front, back, or end, thus greatly facilitating the setting of tool holders and tools. It is of equal importance that an operator, when setting up his tools, should have control of the various machine operations—starting and stopping the spindle, changing the spindle speed, operating the turret slide and the cross slides. These controls are by pushbuttons which are coveniently located on the front of the machine. The open controls are those used normally by the operator in running the machine. Those used in setting-up by hand are under a hinged cover. One switch cuts off all automatic circuits and transposes the machine to manual operation and vice versa.

The flexibility of any automatic machine is greatly increased when the timing of the cross slides can be varied with respect to the main tool slide. With a reversing drive on the cam that controls the cross slide and with retractable cam rolls, an arrangement was worked out whereby only one cam plate with two paths of travel furnished flexible cross slide operation merely by a change in the time of engagement of the cam roll.

The cross slides may be set for "normal" operation, as when the feed stroke ends with the turret feed stroke, or for "late cross slide" operation when cross slides start a stroke only after the turret ends its forward stroke and dwells. This latter operation is for recessing tools, back facing, etc., and either of these operations can be performed on any or all turret faces.

After the operator has set his tools, he must select the desired spindle speeds and feeds, and set the necessary control dogs so that operations will be performed in the proper sequence. To vary his range of automatic spindle speeds, and of automatic feed changes, he simply swings open the covers of the two change gear boxes. The change gears can then be removed simply by sliding them off of their castellated shaft without the necessity of removing nuts or washers, for they are held on by the covers themselves. Electrical interlocks make it impossible to run the machine until these covers are swung back into place.

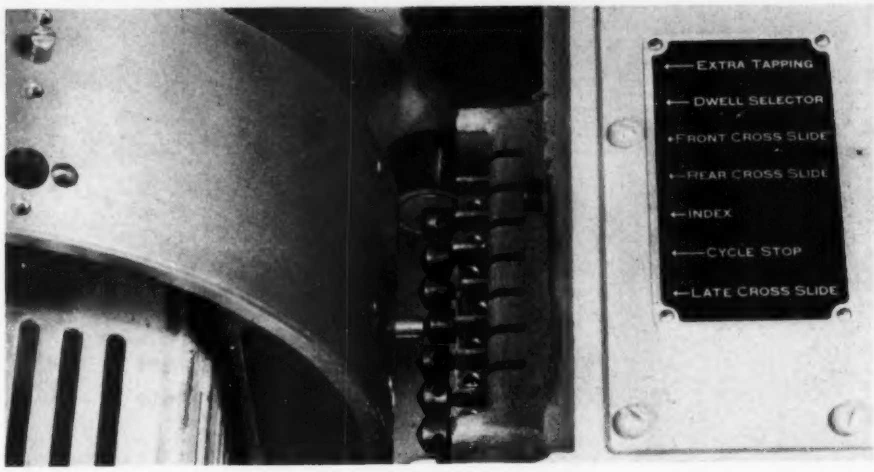
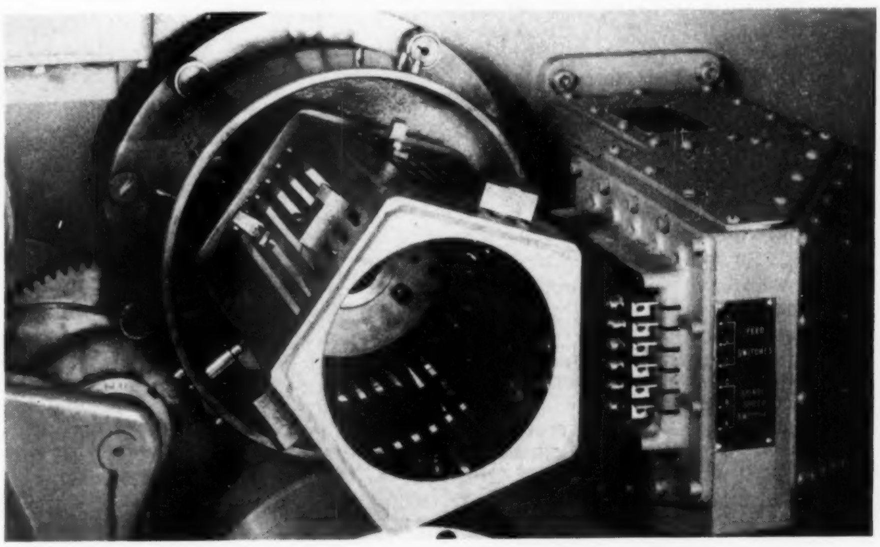


Fig. 2—Above—Switch system for selection of machine functions. Dogs on drum trip switches shown at right

Fig. 3—Below—Pentagon drum system for both selection and positioning of functions



The type of control selected has a direct effect on the reduction of set-up time. When available systems of control were first considered, it was decided that the machine had to have the following characteristics: (1) Complete hand control of all functions and means to quickly change from automatic to hand, and vice versa; (2) Controls to be conveniently located; (3) Control setting elements to be accessible, small in size, and convenient to set.

Based on these specifications, an electrical system was chosen over a mechanical or a hydraulic system. With limit switches for automatic control, more freedom could be obtained in selecting a location for the setting elements. Small dogs or screws could then be used to operate the switches. Hand controls could at the same time be conveniently situated on an operator's panel, thus facilitating the problem of cutting either manual or automatic circuits into operation.

The dogs for operating the switches which control the automatic operation of the machine are located on three drums. Those on the drum at the rear of the machine, *Fig.* 2, operate limit switches for such operations as regulating the dwell of the cutting tool, starting a late cross slide feed, reversing the motion of the slides at the end of the stroke, etc. Adjustments for these operations seldom have to be made. Such functions of the machine as cutting in a cross slide, setting up the machine circuit for skip indexing of the turret, tap-

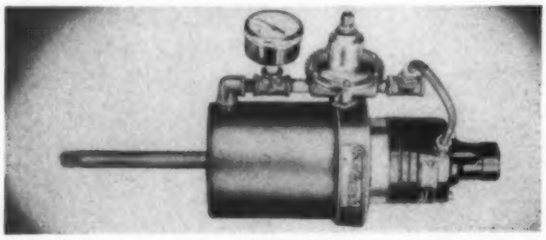
(*Continued on Page* 250)

NEW PARTS AND MATERIALS

. . . presented in quick-reference data sheet form for the convenience of the reader. For additional information on these new developments, see Page 229

Pneumatic Positioner — 1

Conoflow Corp., Philadelphia, Pa.



Designation: B-50 series

Style: Bucket type; integral piston positioner

Size: Bores 4, 6 and 8 in.; strokes to 4 in.; ¼-in. pipe ports

Service: Air to 100 psi

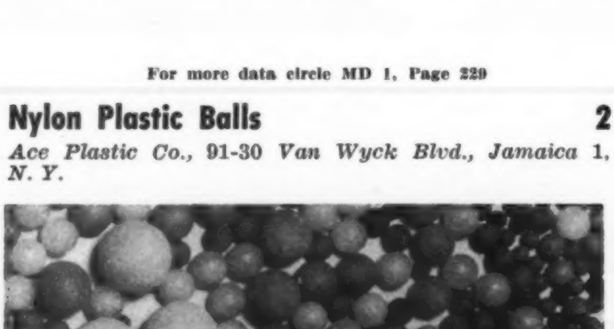
Design: Repositioning accuracy to 1/500th part of total piston travel; cast aluminum cylinder; O-ring seals throughout

Application: For pneumatic systems on machines where accurate positioning control is required, i.e., adjustable-stroke pumps, proportioners, valves, etc.

For more data circle MD 1, Page 229

Nylon Plastic Balls — 2

Ace Plastic Co., 91-30 Van Wyck Blvd., Jamaica 1, *N. Y.*



Style: Seamless molded

Size: ⅛ to ¾-in. diameters

Service: Max operating temp, 480F; coefficient of linear thermal expansion, 5.5×10^{-5}; thermal conductivity 1.7; short time dielectric strength 385; 60-cycle dielectric constant, 4.1; water absorption, 1.5; abrasion and chemical resistant

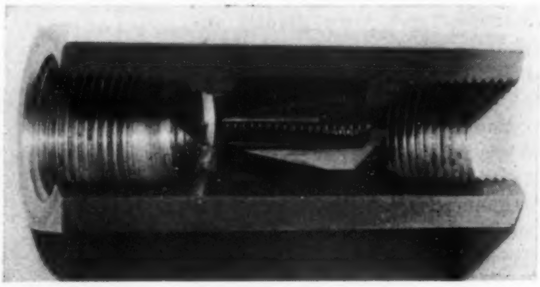
Design: Precision made, ±0.001-in. tolerances

Application: For use as valve components, specialized friction bearings, detents, mechanical checks, etc.

For more data circle MD 2, Page 229

Hydraulic Check Valve — 3

Greer Hydraulics Inc., Brooklyn, N. Y.



Style: In-line; poppet type, spring return

Size: From ½ to 2-in. pipe; special sizes to suit

Service: Hydraulic fluids at pressures to 5000-psi max

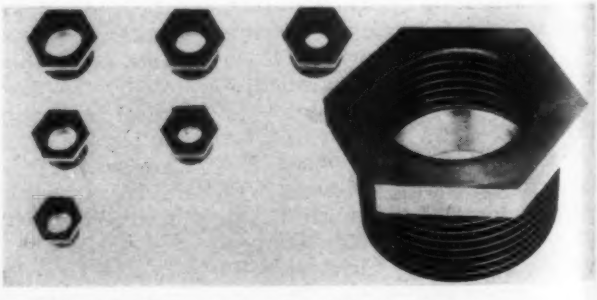
Design: Teardrop shape poppet on hollow shaft with dashpot; fluid flow through calibrated orifice snubs poppet action, preventing shocks and hammer in opening and closing; general design results in low pressure drop

Application: For high-pressure, high-flow hydraulic circuits where normal check valve surges have adverse shock effect on system.

For more data circle MD 3, Page 229

Plastic Pipe Reducers — 4

Carlon Products Corp.; 10225 *Meech Ave.; Cleveland* 5, *O.*



Style: Threaded reducing bushing, hexhead

Size: Fifteen sizes from 2 x 1½-in. to ¾ x ½-in.

Service: For Carlon plastic pipe; all fluids and gases and wide range of chemicals; nontoxic; insulating

Design: Standard bushing with international pipe threads; one piece injection molded thermoplastic; lightweight; smooth internal surfaces do not accumulate scale

Application: For joining sections of plastic pipe or plastic to metallic pipe in low-pressure systems, food processing equipment, insulator conduit, etc.

For more data circle MD 4, Page 229

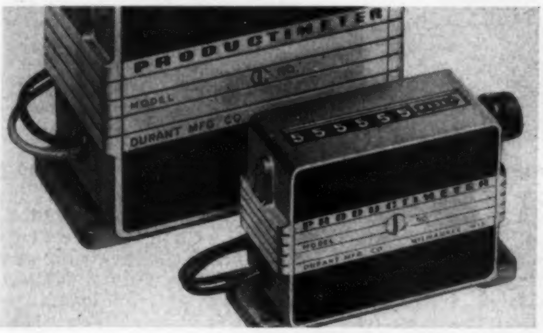
(*Continued on Page* 213)

(Continued from Page 208)

NEW PARTS AND MATERIALS

Electric Counter 5

Durant Mfg. Co., 1933 No. Buffum St., Milwaukee 1, Wis.



Designation: Productimeter; Model 6-Y-1-MF

Style: 6-digit, manual reset; panel or base mount; totally enclosed

Size: 1⅜ w x 2-9/16 h x 3¾-in. long; figures 3/16 high x 0.138-in. wide

Service: Light-duty; 1000 counts per minute; 110/220v 60 cycle a-c

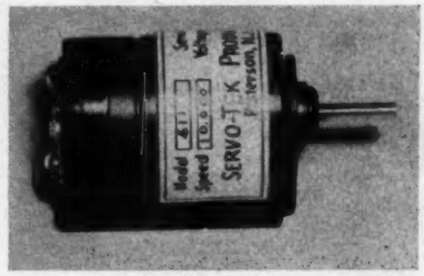
Design: Hardened steel working parts; die cast zinc frame, aluminum side covers; baked enamel finish

Application: For electrically actuated counting in repetitive processes where dial is read at close range.

For more data circle MD 5, Page 229

Miniature D-C Motor 7

Servo-Tek Products Co., Paterson, N. J.



Designation: Type 61 Series

Style: Permanent magnet type; face or base mount; totally enclosed; type 611 without face mount holes, type 612 with face holes

Size: 1/1000 to 1/200-hp; 1 7/64-in. diameter; 2 1/32-in. long including ½-in. shaft extension (without base), 2½-in. long with base; 0.120-in. shaft diameter; weight 2.47 oz (without base), 2.74 oz (with base); face mounting—3 #2-56 tapped holes on 0.875 diameter hole circle; base mount 4 #4-48 tapped holes

Service: Voltage ratings 6 to 28v d-c; type 611,27v motor—max output 4.9 watts at 6500 rpm, 1.0 oz-in. torque—max efficiency 53% at 10,500 rpm, 0.3 oz-in. torque

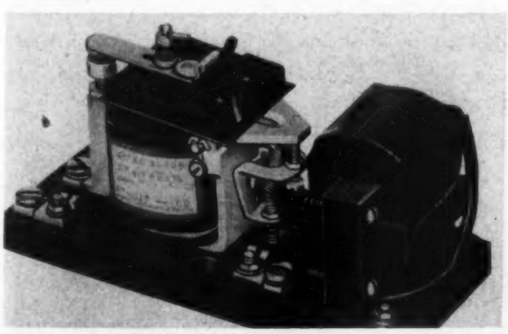
Design: Cylindrical Alnico V-field magnet in conjunction with 14-commutator segment armature; precision ball bearings; high-altitude brushes

Application: Drive for high-altitude aircraft equipment and special industrial applications such as leak detectors, etc.

For more data circle MD 7, Page 229

Power Relay 6

Ward Leonard Electric Co., Mt. Vernon, N. Y.



Designation: Bulletin 401

Style: Voltage sensitive, magnetic, SPNO; 6510, 7510 and 8510 (heavy duty); 6515 and 7515 (light duty); open or enclosed; front mount

Size: Light duty—3⅞ x 6⅜ x 3⅝-in. deep (enclosed), 2¼ x 4⅝ x 2-in. (open); heavy duty—5⅝ x 8½ x 4⅛-in. deep (enclosed), 2½ x 6 x 3 1/16-in. (open);

Service: Heavy duty—(6510) 115v 60 cycle a-c, 105v pick-up, 95v drop-out, 25 amps max noninductive load, and 1 hp; (7510) 230v 60 cycle, 205, 185, 25, and 2; (8510) 440v 60 cycle, 400, 365, 15, and 1—light duty—(6515) 115v 60 cycle a-c, 105v pick-up, 95v drop-out, 25 amps max noninductive load, and ¼-hp; (7515) 230v 60 cycle, 205, 185, 15, and ½

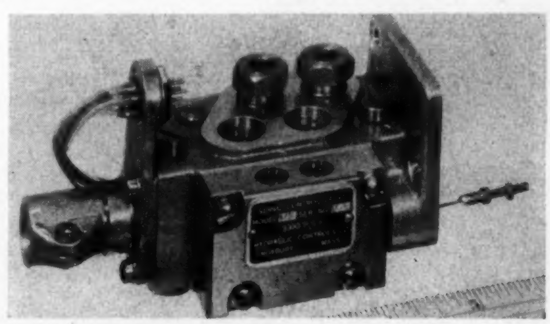
Design: Silver, self-aligning, self-cleaning contacts; chatter-free; coil and reactor connected in series

Application: For a-c circuits where close voltage differential between relay pick-up and drop-out is needed; protection of a-c motors to 2 hp.

For more data circle MD 6, Page 229

Servo Control Valve 8

Hydraulic Control Co., Roxbury 20, Mass.



Designation: Models MX and XA

Style: Pilot valve and power valve as unit

Size: 2⅜ x 2 11/16 x 4 9/16-in.; tubing 5/16-in. (MX), ¼-in. (XA)

Service: Hydraulic fluid to 3000 psi max; max power output 5 hp (MX) 2½ hp (XA); max output flow rate 8.8 gpm (MX), 4.1 gpm (XA); pilot valve max stroking force 135 lb/in. (MX) 75 lb/in. (XA)

Design: Pilot valve requires electromechanical transducer (i.e., torque motor) with force output of 4 lb for 0.004-in. piston displacement; position of power valve measured electrically by linear differential transformer pickoff accurate within 1% over range of power valve travel; transformer pickoff sensitivity 9v per in. when excited with 10v 400 cycle; 0.0002 to 0.0004-in. underlap between piston lands and ports for low dither level and continuous linear output to displacements of 0.0001-in.; internal leakage at 3000 psi 0.15 gpm in each valve

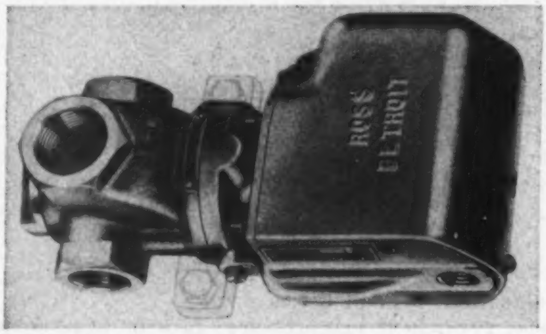
Application: For high-performance electrohydraulic servomechanisms requiring high natural frequencies.

For more data circle MD 8, Page 229

NEW PARTS AND MATERIALS

Air Control Valve 9

Ross Operating Valve Co., Detroit 3, Mich.



Style: Straightway or 3-way, normally open or closed; in-line or bracket mounted

Size: ¼-in. taper pipe ports (8⅞ h x 5 w x 3⅝-in. deep, to 1¼-in. taper pipe ports (8⅞ h x 5 w x 5 1/16-in. deep)

Service: Air line pressures 40 to 125 psi; current consumption at 115v 60 cycle, 0.56 amps holding, 4.3 amps inrush

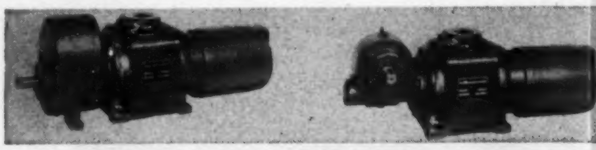
Design: Mounts in inverted position only; valve returns to initial position if air or electric power fail; major pistons and poppets in single unit to insure simultaneous action; oversize exhaust for fast control; pilot lever floats on needle bearings; clapper type solenoid armature floats on rubber cushion; no springs; dust-tight solenoid cover hinged for access, opening in cover permits manual operation; corrosion resistant parts, molded Hycar poppet seats

Application: For control in machine air circuits where valve, in event of power interruption, must automatically assume normal position for safety.

For more data circle MD 9, Page 229

Variable-Speed Transmission 11

Graham Transmissions Inc., 3754 N. Holton St., Milwaukee 12, Wis.



Designation: 150 Series, Models 150, 175, 200 and 250

Style: Straight line extension of induction motor; built-in motor or shaft extension for coupled motor or offset drive, built-in single stage spur reduction, built-in worm reduction, built-in helical spur reduction; direct or remote control; standard NEMA C flange motors of all types—open, explosion proof and totally enclosed

Size: ½ hp (Model 150), ¾ (175), 1 (200) and 1½ (250)

Service: Continuous heavy-duty; input speeds 1800 and 1200 rpm; standard construction, output from 0.37 input to 0 rpm

Design: Spring-loading eliminated to provide for speed change while motor is running or stationary; lever or single-turn friction lock control for speed holding; overload protection

Application: For drive on small machine tools, pumps, contact printers, winders, etc., where low cost is imperative.

For more data circle MD 11, Page 229

High-Speed Motor 10

John Oster Mfg. Co., Racine, Wis.

Designation: HC-303-Model 1, HC-268-Model 2

Style: Capacitor run single-phase or polyphase induction motor; totally enclosed; face mount; Model 2 fan-cooled

Size: 1/100-hp; 1 13/16 in. housing diameter; 2¼-in. diameter over fan shroud Model 2; lengths including 17/32-in. shaft extension—3 25/32 in. (Model 2), 3⅜-in. (Model 1); shaft diameter 3/16-in., max ¼-in.; mounting No. 6-32 NC 2, 3 holes on 1.281-in. bolt circle

Service: Continuous duty with fan; 7200 rpm; 400 cycle 115v a-c, 0.25 amps, 25 watts; condenser, 0.6 mfd, 220v a-c; ambient temp range 55 to 70 C

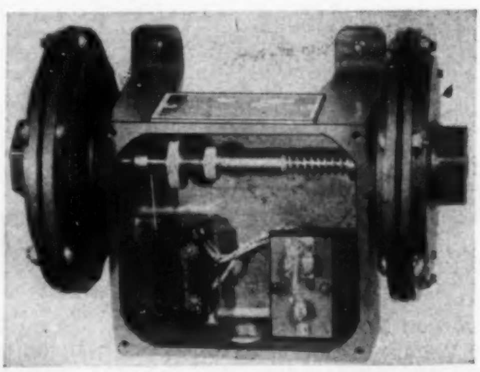
Design: External fan and shroud can be made with demountable construction for ease of replacement of windings or in a permanently sealed construction for maximum moisture resistance; 6-in. leads stripped and tinned ¼-in.

Application: High-speed, high-power drives where size and weight must be minimum.

For more data circle MD 10, Page 229

Pressure or Vacuum Control 12

Coral Designs, Div. Henry G. Dietz Co., 12-16 Astoria Blvd., Long Island City 2, N. Y.



Designation: Cat. 111-D

Style: Diaphragm type; SPDT (NO or NC) switch

Size: 8½ w x 7 h x 5-in. deep

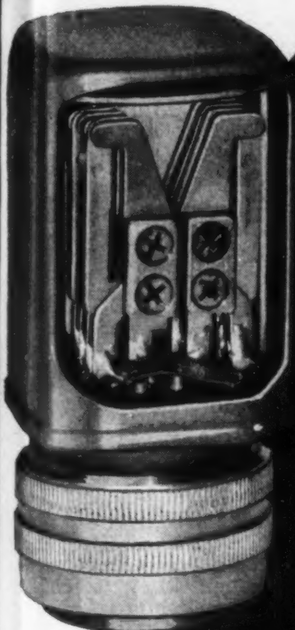
Service: Pressure or vacuum control; standard ratings to 20-in. water or vacuum, differentials from 0.2 to 2-in. water; UL ratings 10 amps 125v or 5 amps 250v

Design: Two neoprene rubber diaphragms, each with one side to pressure and other to atmosphere; individual diaphragms for each vessel to reduce leak to minimum; disks, one at each end of shaft, actuate Micro Switch; adjustment of range by spring-loading shaft; cast aluminum housing; cadmium plated brass machined parts

Application: For regulating pressure between two vessels where no leak path is desired; direct start-stop circuit without relays.

For more data circle MD 12, Page 229

AN approved (3303-1)



R-B-M

ELECTRONIC AND COMMUNICATION RELAYS

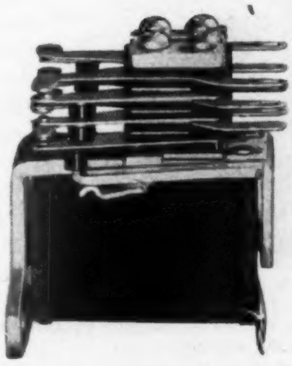
Now Hermetically Sealed

AN approved (3304-1)

● Hundreds of thousands of R-B-M telephone type relays saw Government service in World War II. Now most of these relays are available in hermetically sealed enclosures designed to meet AN specifications.

R-B-M hermetically sealed telephone type relays are available in contact forms up to and including 4-pole, double throw, 3 ampere, 28 Volts D.C. construction. Also 10 ampere rating up to and including 2-pole double throw at 28 Volts D.C. All relays available with approved AN plug connector, or solder connections.

Engineers! What is YOUR hermetically sealed relay requirement? R-B-M is developing new and smaller relays to meet Armed Services requirements. Perhaps one of these will solve your problems. Write giving complete relay specifications, application, quantity and AN specifications applying. Address Dept. B-4.



R-B-M Production and Engineering facilities in two plants, located in different states, (over a quarter million square feet), can assist you in the development and production of special electro-magnetic devices for Armed Services application.

RBM

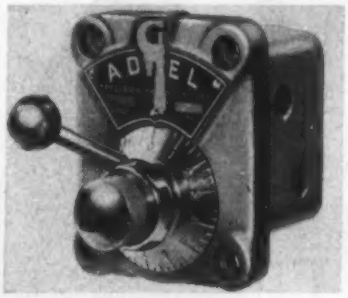
R-B-M DIVISION ESSEX WIRE CORP.
Logansport, Indiana

MANUAL AND MAGNETIC ELECTRIC CONTROLS — FOR AUTOMOTIVE, INDUSTRIAL, COMMUNICATION AND ELECTRONIC USE

NEW PARTS AND MATERIALS

Hydraulic Valve 13

Adel Div. General Motors Corp., 10777 Van Owen St., Burbank, Calif.



Style: Flow-control; manual adjustment

Size: ⅜ and ¾-in. pipe ports

Service: Hydraulic fluids to 1500 inlet psi max; 1, 2, 4 and 6 gpm (⅜-in); 10, 15, 20 and 25 gpm (¾-in.)

Design: Pressure compensated for constant flow; adjustable from complete shutoff to max flow; max recommended outlet pressure equals inlet pressure minus 100 psi; 270-degree control lever rotation; no drain line required

Application: For machine hydraulic circuits where constant fluid flow and constant machine feed are desired over wide range of pressures and feed resistances.

For more data circle MD 13, Page 229

Timing Belt 15

L. H. Gilmer, Div. United States Rubber Co., Rockefeller Center, New York

Style: Flat belt with teeth, endless

Size: Heavy-duty 27 to 140-in. long x 1 to 3-in. wide; light-duty 21 to 54-in. long x ½ to 1-in. wide; special sizes to suit; pitches 0.500-in. (heavy) 0.375-in. (light)

Service: To 16,000 fpm; 53 hp at 11,000 fpm (1-in. heavy); 15 hp at 8,000 fpm (1-in. light); speed ratios to 30:1

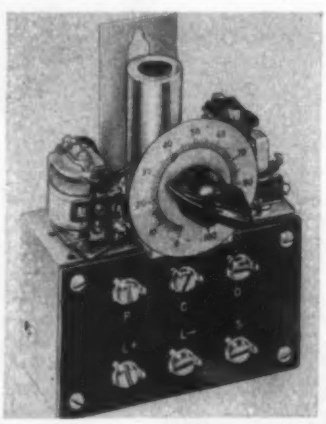
Design: Neoprene or Buna-N rubber with steel wire core (heavy-duty) and Ustex cotton cord (light); steel core permits operation on fixed centers without adjustment, eliminates elongation, gives high tensile strength; low noise level; no lubrication required; flexibility permits pulley diameters as small as ½-in. at 10,000 rpm; sensitive to gross misalignment

Application: For machine drives where nonslip, precise timing transmission of power is desired; color television drive mechanism, sewing machines, automatic screw machines, pumps, portable power equipment, etc.

For more data circle MD 15, Page 229

Electronic Timer 14

Farmer Electric Co., 21 Mossfield Rd., Waban, Mass.



Designation: Type CK

Style: Cold-cathode tube; model A surface mount, model B panel mount; manual setting

Size: Model A 3⅞ x 4⅞ x 3 in. deep; model B 3⅞ x 4⅜ x 2 in. deep

Service: Time ranges 1.5, 3, 6, or 12 seconds max; dial with 100 graduations; load contact rating 8 amps noninductive at 125v a-c; input 105-125v 60 cycle

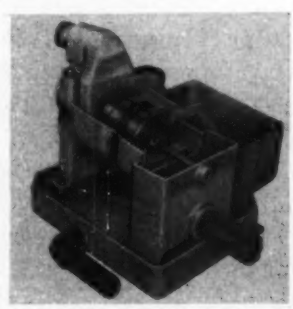
Design: No standby current required by cold cathode; no recharging time required by condenser before start of next cycle; external loads connected to terminals without affecting timer; load contacts SPDT

Application: For short-interval time control of 0.03 to 12 seconds on such processes as spot welding, heat sealing and conveying.

For more data circle MD 14, Page 229

Coolant Clarifier 16

Honan-Crane Corp., Lebanon, Ind.



Style: Package unit; magnetic; open; forced or gravity flow

Size: 15 3/16-in. w x 16½ h x 22⅝ in. long, net weight 145 lb; inlet and outlet connections 2-in. NPT; overflow 1-in. NPT on 2 sides; mounting hole dimensions 20⅛ x 12 13/16-in.

Service: Continuous clarification of liquid coolants (mineral oils and soluble oil solutions) to 20 gpm max; 1/12-hp motor, 60 cycle 110v single phase

Design: Automatic; permanent ring magnets completely enclosed in revolving nonmagnetic cylinder; circular steel disks, endpieces of cylinder, act as magnetic poles to seal magnet cores from fluid; 360-degree attraction; nonmagnetic wiper trough for nonmagnetic abrasive particles; cycling pump or separate pump and motor force flow through unit (special interchangeable inlet castings used for gravity flow; overage returned to sump

Application: For wet surface grinders, thread grinders, milling machines, gear shapers, etc., where continuous flow of clean lubricant is desired.

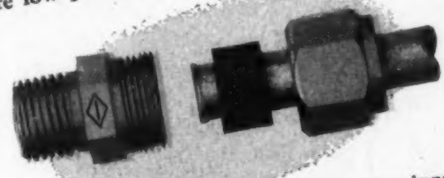
For more data circle MD 16, Page 229

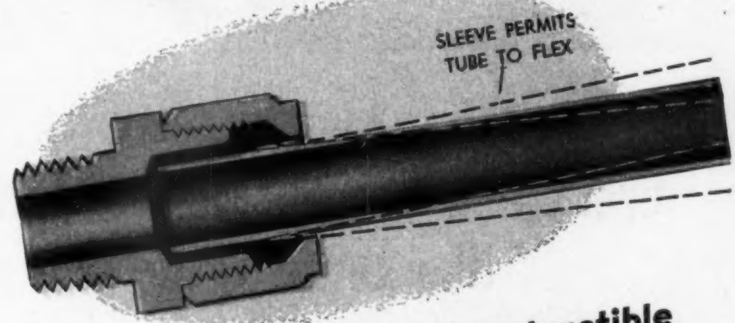
If You Connect Tubing You Should Know About . . .

IMPERIAL FLEX FITTINGS

. . . the tube coupling with the vibration and shock absorbing sleeve

Easy to Install . . . All that is necessary is to slip nut and Flex sleeve over tubing. Then insert tubing into fitting body as far as it will go, and assemble. This applies to sizes ½" O.D. and smaller where low pressures are involved.





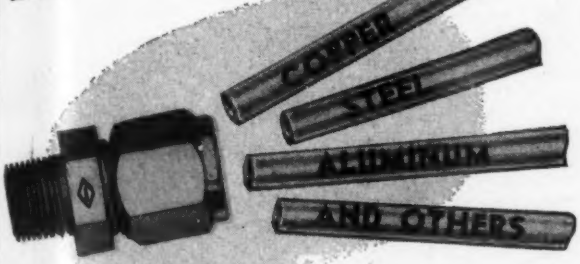
On sizes larger than ½" O.D. and on applications where higher pressures are involved, end of tube should be belled slightly, as shown in illustration above.

Makes Joints Virtually Indestructible by Vibration . . . . The Flex Fitting embodies a sleeve of special synthetic elastic material which permits the tubing to flex back and forth through the angle shown and at the same time assures a positive, pressure-tight seal.

It assures a safe, durable tubing installation that will withstand major vibration, minor tube movement and considerable mechanical shock without damage to either tubing or fittings. On tests where ordinary fittings failed after 73,000 cycles of vibration, Imperial Flex Fittings have withstood over 20,000,000 cycles without failure.

Flex Fittings eliminate the need for costly flexible hose lines in many instances.

Proved by extensive service in the field. Used as standard equipment on Diesel engines, oil filter connections, heavy power equipment, machinery, etc.



For All Kinds of Tubing . . . . . . Imperial Flex Fittings can be used with all types of seamed and seamless metal tubing, including copper, aluminum, thin-wall steel, Monel, stainless steel, etc.

Catalog No. 344 gives complete engineering data on Flex Fittings including types, sizes, dimensions, specifications and application information. **Write for your copy.**

Emblem of Quality

IMPERIAL

THE IMPERIAL BRASS MANUFACTURING COMPANY

513 Racine Ave., Chicago 7, Illinois

PIONEERS IN TUBE FITTINGS AND TUBE WORKING TOOLS

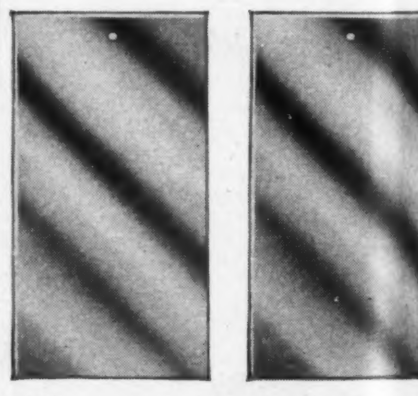
ADVERTISEMENT

Clear Coatings for Zinc and Steel Stand 800 hours' Salt Spray

Increased use of zinc die castings and of zinc-plated steel to replace unavailable materials, combined with the tight supply of copper, nickel and chromium normally used for plating zinc, has focused attention on surface coatings comparable to plating in service performance. Unbiased laboratory tests show that at least two of the clear finishes in the company's line withstand the exceptionally long period of 800 hours' exposure to salt spray and to weatherometer tests.

Effectively protect zinc

These tests demonstrate that DULAC Clear Universal Lacquer #462 and CODUR Clear Synthetic Y743 provide completely satisfactory protection on zinc, zinc plated steel and steel. Even after the unusually severe tests to which these finishes were subjected, there was no indication whatever either of failure of the coating or of discoloration of the zinc.



(Left) A zinc-plated steel panel newly coated with DULAC #462. (Right) A similar panel after 800 hours' exposure to salt spray, showing no evidence of attack on the finish.

Adaptability to Drying Schedules

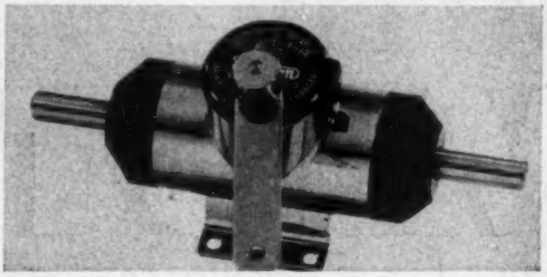
While both finishes give the same performance, DULAC #462 is an air-drying coating, while CODUR Y743 is a baking type. This permits choice of the correct finish to fit into the drying schedules of a particular finishing room.

Technical Data Bulletin #110 on clear finishes is available from Maas & Waldstein Co., 430 Riverside Avenue, Newark 4, N. J. On request, M & W Technical Service Engineers will discuss specific problems.

NEW PARTS AND MATERIALS

Miniature Variable-Speed Drive 17

Metron Instrument Co.; 432 Lincoln St.; Denver 9, Colo.



Designation: Type 4B

Style: Roller and disk; lever speed control

Size: 0.025 hp; 2 lb-in. torque

Service: 20,000 rpm max; ratio infinitely variable from 1/6 to 6

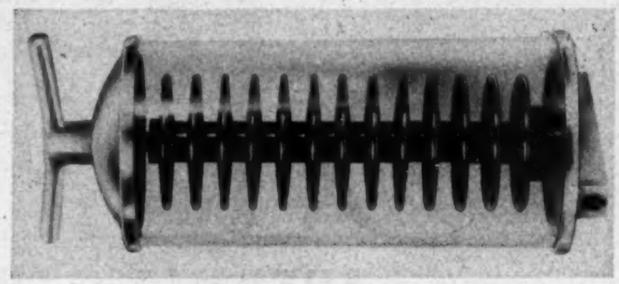
Design: Compact sealed construction; speed adjusted by changing radius at which rollers make contact with disks, rollers directly turned by lever control; actuating lever has pointer which rides on calibrated ratio scale; automatic or remote control possible with linkages or cams

Application: Drives for timers, computers, business machines and similar low-power devices where manual, automatic or remote control of speed is desired.

For more data circle MD 17, Page 229

Glass-Walled Filter 19

Micro Metallic Corp., 30 Sea Cliff Ave., Glen Cove, N. Y.



Designation: Surfamax and Gravitain

Style: Filter elements in cylindrical glass containers

Size: To suit, 4½-in. diameter x 40-in. long container accommodates 8 sq. ft. of filter area

Service: Filters particles from 5 (0.0002-in.) to 165 microns (0.0065-in.) minimum; pressures to 40 psi, integral stainless steel guard screen for higher pressure

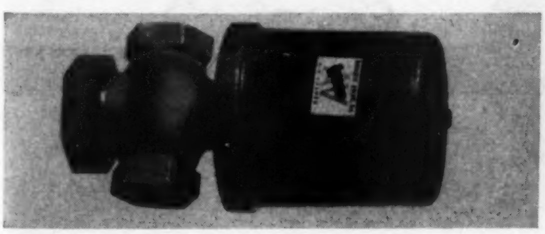
Design: Filter septum is porous stainless steel sheet with 50 per cent voids; types 304 and 316 stainless and Hastelloy B used in construction, other corrosion resistant alloys special; Pyrex glass container for heat and pressure resistance

Application: For commercial filtering where the influent suspensions have variable properties and in research and development work.

For more data circle MD 19, Page 229

Fluid Shut-Off Valve 18

Airmatic Valve Inc.; 1643 E. 40th St., Cleveland, O.



Designation: Model LDS-2-750

Style: In-line, direct solenoid, capacity shut-off

Size: ¼ through 2-in. standard pipe ports

Service: Oil, water or air; pressures to 225 psi max; solenoid a-c or d-c

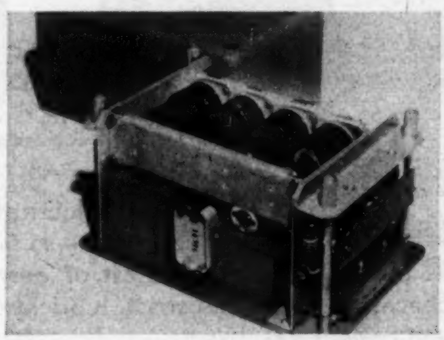
Design: One moving part; corrosion-resistant construction, cast naval bronze; positive seal; mounts in any position; inspection and servicing without disturbing piping

Application: For automatic shut-off control of fluid pressure systems operating equipment and machine processes; particularly adapted to restricted space installation.

For more data circle MD 18, Page 229

Servo Amplifier 20

Servomechanisms Inc., Mineola, N. Y.



Designation: SA-203

Style: Packaged system, plug-in, panel mount

Size: 5 x 8 x 4⅜ in. high; 5½ lb

Service: Output 115v 60 cycle 15 watts max (tuned for control phase of Diehl FFE-25-11 or equivalent motor); input impedence 1 megohm (without network NA); with NR102 damping, net gain is 4000; error rate damping to suit; carrier signals of 60± 2 cycles per second; operates at ambient temperatures to 140F

Design: Amplifier contains transmission control plug-in networks for input (NA), phase shift (NP) and damping (NR)—three stages of voltage amplification—one driver stage—and output power stage, transformer coupled to motor; load conditions and data devices changed by exchange of net-works

Application: For use in high-performance velocity or position servo loops in such as analogue computing devices, laboratory measuring instruments, automatic processing and inspection controls, etc.

For more data circle MD 20, Page 229

(Continued on Page 222)

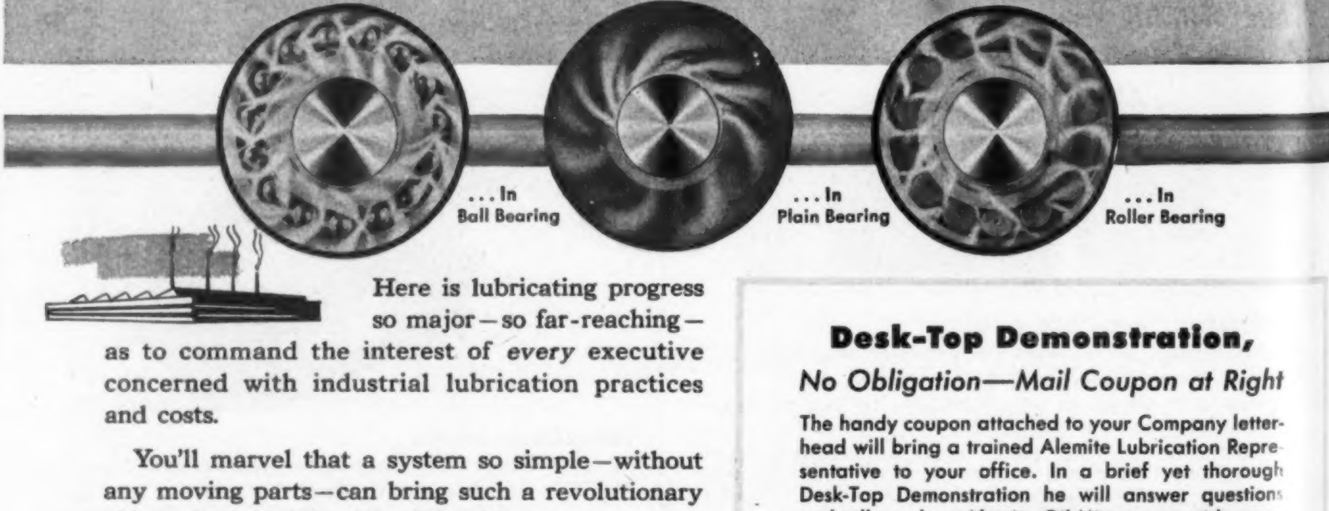
Alemite Announces THE MOST AMAZIN

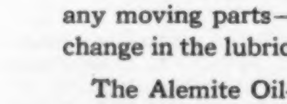
ALEMITE OIL-MIST

AUTOMATIC LUBRICATION SYSTEM

Unbelievably simple system atomizes oil into mist, circulates it through tubing to bearings. Bathes all bearing surfaces with fresh, clean, cool oil film. Uniformly maintains oil film on all sliding, rubbing, rolling parts regardless of variations in load, temperature or speed! No "peaks and valleys" of lubrication.

Fully automatic—eliminates waste and the uncertainties of the "human element." Extends bearing life as much as 17½ times. Seals bearings against dirt and abrasives. Cuts oil consumption as much as 90%. Greatly reduces the number of oils needed.





Here is lubricating progress so major—so far-reaching—as to command the interest of *every* executive concerned with industrial lubrication practices and costs.

You'll marvel that a system so simple—without any moving parts—can bring such a revolutionary change in the lubrication of machinery.

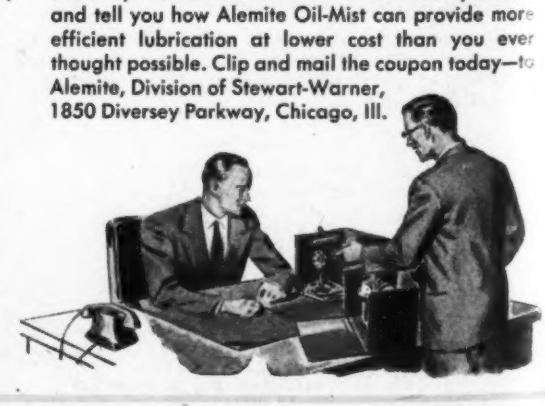
The Alemite Oil-Mist System requires only two simple settings—to control the amount of air pressure, and to regulate the density of the Oil-Mist. Once set, they require no further attention.

This great new Alemite Oil-Mist System has been proved by plant installations on a wide variety of machines in many industries including steel, coal, food processing, chemicals, metalworking, textiles and woodworking. The results are spectacular—in reduced lubrication costs, reduced maintenance costs, reduced service, reduced "down-time."

Desk-Top Demonstration,

No Obligation—Mail Coupon at Right

The handy coupon attached to your Company letterhead will bring a trained Alemite Lubrication Representative to your office. In a brief yet thorough Desk-Top Demonstration he will answer questions and tell you how Alemite Oil-Mist can provide more efficient lubrication at lower cost than you ever thought possible. Clip and mail the coupon today—to Alemite, Division of Stewart-Warner, 1850 Diversey Parkway, Chicago, Ill.

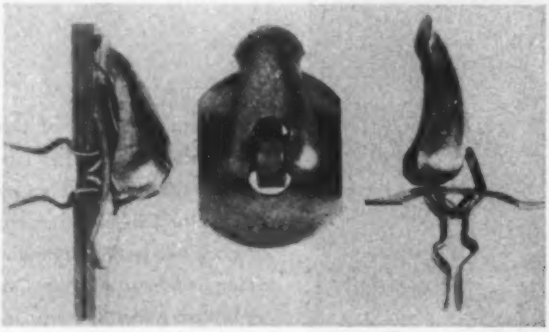


(Continued from Page 219)

NEW PARTS AND MATERIALS

Locking Fastener 21

American Shower Door Co., Los Angeles, Calif.



Designation: Thumweld

Style: Quick insert, snap locking

Size: Hole sizes from ⅛ to ½-in.

Service: Metal thicknesses to ½-in., variations of −0.005 to +0.020-in.; 15 lb load per fastener (3/16-in. size)—where vibration is encountered, 5 lb per fastener

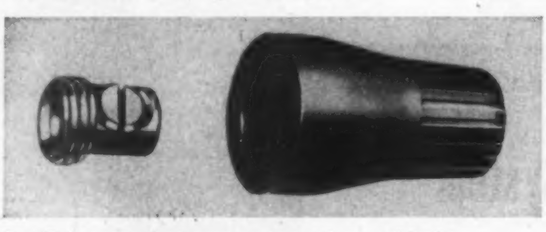
Design: Three-piece fastener: Anchor pin, sheet metal washer with v-projection, and locking lever; pressure on lever pulls anchor pin upward against v and spreads anchor points which grip metal being fastened

Application: For permanent metal fastening where rapid assembly or disassembly is important, or to replace screws or bolts on units such as access plates, metal housings, panels, instruments, etc.

For more data circle MD 21, Page 229

Solderless Wire Connector 22

Ideal Industries Inc., Sycamore, Ill.



Style: Setscrew pressure connector, solderless

Size: Two sizes, Nos. 11 and 22

Service: UL approved for 600 v; wire combinations—(No. 11) 2 #12, 1 #12 and 1 or 2 #14, 2 or 3 #14—(No. 22) 2 #10, 2 #10 and 1 #12 or #14, 2 or 3 #12, 1 or 2 #12 and 1 or 2 #14, 1 #12 and 1 to 4 #14, 2 to 5 #14

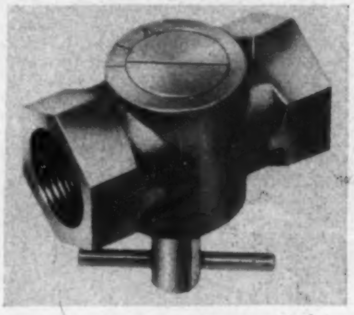
Design: Solid brass setscrew and sleeve; phenolic shell screws over sleeve and can be easily removed for inspection of joint or for circuit remake; screw driver pressure insures permanent joint

Application: For general use in branch circuit wiring, fixture hanging, machine and appliance hook-up where a permanent solderless splice of two or more wires is desired; ideal for use where circuit layouts are planned for possible addition of new circuits.

For more data circle MD 22, Page 229

Regulating Valve 23

Reynolds Shaffer Co., Detroit 4, Mich.



Designation: R-V1 (full ported) R-V2 (small port for fine adjustment)

Style: Stop cock; manual adjustment

Size: 2½ in. long; ¾-in. pipe; R-V1 has 1-in. pipe area port in plug, R-V2 has ¼-in. plug port; larger sizes to suit

Service: Water and other fluids; exact flow regulation

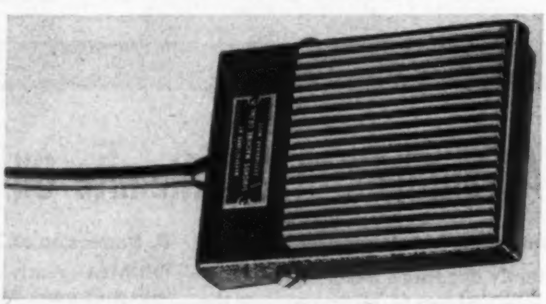
Design: All brass, close fitted plug (Bearenized finish on plug and ID to 0.0005 in.); positive O-ring packings; quick disassembly; one-quarter turn opens or closes

Application: Used as a flow-regulating valve for jet and spray pumps, by-pass valve on feed lines, etc.

For more data circle MD 23, Page 229

Miniature Foot-Operated Switch 24

Simonds Machine Co. Inc., Southbridge, Mass.



Designation: Linemaster Treadlite

Style: T-51-S (one contact) and T-52-S (two contacts); SPDT (NO,NC), foot-operated

Size: 3½ x 2⅝ x 1 in.

Service: 5 amp at 110/250v, momentary contact

Design: T-52-S operates two circuits—partial pressure on treadle actuates one contact, further pressure a second contact; formed-steel casing with black crackle finish

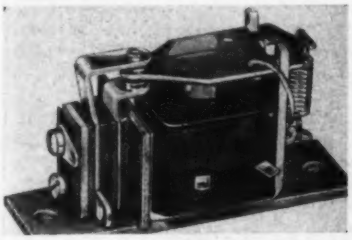
Application: For foot-operated control of low-voltage light-duty electrical circuits of business machines, sound equipment, magnetic switches, solenoids, relays, etc.

For more data circle MD 24, Page 229

NEW PARTS

Plate Circuit Relay 25

Potter & Brumfield, Princeton, Ind.



Style: LS series, SPDT

Size: 2⅝ x 1¼ x 1 5/16 in. high

Service: 2500, 5000 and 10,000 ohm coil resistances; contacts 5 amp; min actuating power 0.09 w, normal operating power 0.2 w

Design: Open construction; designed for min size and low cost; adjusted to pull in at approximately 5 ma

Application: For photoelectric control circuits and electronic timing devices.

For more data circle MD 25, Page 229

Vinyl Plastic Coating 26

Casey & Case Coating Co., P. O. Box 151, Maywood, Calif.

Form: Inert thermoplastic resins in formulation for prime, intermediate, and finish coats; colors—gray (prime) white, black, gray and oxide red (intermediate), white, black, gray, oxide red, cream, green, and aluminum (finish)

Service: Exterior and interior coatings for severe corrosion or contamination environments; max temperatures 160F; applied by brush, spray or dip

Properties: Generally resistant to acids, alkalies and salts except acetic acid (over 2%) formic acid, hydrofluoric acid, chlorinated hydrocarbons, aromatic hydrocarbons, aldehydes, bromine, esters, ethers, cresols, carbon disulphide, betons, phenol, and iodine; abrasion resistant; nontoxic; nonflammable

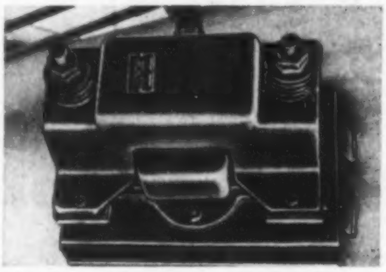
Application: Protective coatings for food processing equipment, chemical handling and storage equipment, marine equipment, and production machines in general.

For more data circle MD 26, Page 229

NEW PARTS AND MATERIALS

Hopper Vibrator 27

Syntron Co., Homer City, Pa.



Designation: V-30, V-60, V-80 and V-240

Style: Pulsating magnet with wall-mounted VCR variable-power control

Size: Weight of vibrator 30 lb (V-30), 43 lb (V-60) 115 lb (V-80) and 247 lb (V-240)

Service: 3600 vibrations per minute; V-30 will vibrate hoppers made of 3/16-in. sheet containing not more than 20 cu ft of material, V-60 ¼-in. and 50 cu ft, V-80 ⅜-in. and 20 ton; input watts 125, 175, 300, and 850 respectively; 110, 220 or 400v 60 cycle a-c

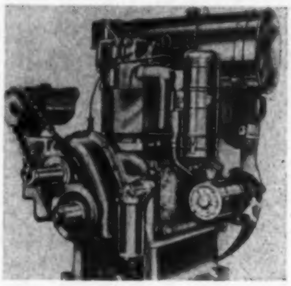
Design: Springs pound against rubber bumpers for "cushion action" and reduced noise; weight increase over standard models to compensate for power loss due to absorption by rubber; electronic controller valve changes a-c to pulsating waves with time interval between resulting in long stroke; controller dial switch varies power

Application: For vibrating bins, hoppers and chutes for free material flow, vibrating forms to eliminate pockets and voids, and vibrating screens to increase capacity.

For more data circle MD 27, Page 229

Diesel Engine 29

Lister-Blackstone Inc., 420 *Lexington Ave., New York 17, N. Y.*



Designation: Model AS 2

Style: Two-cylinder, four-cycle; direct injection

Size: 20 hp; 43½ in. deep x 36⅞ in. high overall; net weight 725 lb

Service: Air-cooled; 1800 rpm; fuel consumption No. 2 diesel oil, 0.38 lb/bhp/hr

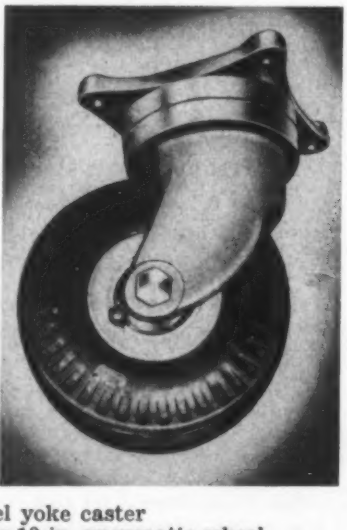
Design: Bore and stroke 4¼ in.; compression ratio 15:1; fuel tank capacity 6.6 gal.; luboil capacity SAE 30, 2.1 gal.; flywheel housing drilled to suit No. 4 SAE pitch circle diameters for standard clutch power take-off and standard reverse and reduction gears

Application: Prime mover for stationary units such as generator and pumping sets and mobile units such as concrete mixers, haybalers, etc.

For more data circle MD 29, Page 229

Pneumatic Tire Caster 28

Aerol Co. Inc., 2820 *Ontario St., Burbank, Calif.*



Style: Swivel yoke caster

Size: Mounts 10-in. pneumatic wheel

Service: Load rating 1200 lb

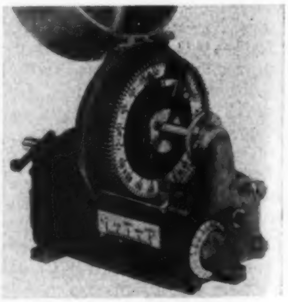
Design: Sparkproof; high rollability to decrease friction heat; cased bearings to minimize race wear

Application: For mobile materials handling equipment used in arsenals; explosives processing plants, etc., where max loads must be moved with minimum vibration and shock.

For more data circle MD 28, Page 229

Predetermined Electric Count Control 30

Counter and Control Corp., 5225 *W. Electric Ave., Milwaukee* 14, *Wis.*



Designation: Cyclo-Monitor, Models CMR and CMS

Style: Rotary (CMR) and stroke (CMS), mechanical; automatic reset

Size: Overall 7⅝ h x 5 13/16 w x 9⅞-in. long including 1⅜-in. shaft extension; ½-in. shaft diameter; mounting by 3 holes ¼-in. diameter

Service: Electric cycle or interval control by count or rotation; to 5000 counts per minute (rpm) max (CMR) to 1000 strokes per minute max (CMS); switching interval ½ to 6 counts; snap-action switch NO or NC, 10, 5 or 3 amps. at 125, 250 and 460 v a-c respectively

Design: One count per revolution (CMR); 60-degree movement of spring-loaded ratchet type lever one count (CMS); when mechanical forces are insufficient to drive, CMS can be operated by 110v 60 cycle solenoid to 350 counts per minute, max; count set by moving pointer on dial gear and turning venier knob; dial gear rotates one way to selected count, actuates switch (interval) and repeats in opposite direction

Application: For cycle control of motors, relays, solenoids, signals, etc., used in electrical machine circuits where actuation of interrelated functions are dependent upon specified numbers of pieces or motions.

For more data circle MD 30, Page 229

Steel Tubing — 31

Carpenter Steel Co., Alloy Tube Div., Union, N. J.

Form: Round welded tube or pipe; alloys B and C of Hastelloy strip

Size: Round tubing from ⅝ through 4½ in.; round pipe ⅜ through 4 in. ips (schedules 5 and 10—light and medium weight), ⅜ through 1 in. (schedule 40—heavy weight); max length of finished tubing with circumferential splices, 22 ft —without, 15 ft (⅝ to 1½ round), 10 ft (over 1½ round); gage 0.035/0.083-in. (⅝), 0.035/0.095 (¾), 0.035/0.019 (⅞ to 1¼) 0.035/0.135 (1¼ to 1½), 0.065/0.134 (1½ to 4½)

Service: Piping systems, especially chemical processing, handling corrosive or oxidizing agents

Properties: Alloy B—resists hydrochloric acid in all concentrations and at all temperatures even boiling; resists boiling sulphuric as well as concentrations above 30% at low temperatures; recommended for hydrochloric acid above 80 C; not satisfactory for oxidizing atmospheres—Alloy C—because of chromium content resists nitric acid, free chlorine and acid solutions of ferric and cupric salts; resists phosphoric acid and acetic, formic, and sulphuric acids; recommended for hydrochloric acid to 50 C—Alloys B and C can be fabricated for bending.

For more data circle MD 31, Page 229

Protective Coating — 32

American Sand-Banum Co. Inc., 9 Rockefeller Plaza, New York 20, N. Y.

Designation: Tinallium

Form: Nondrying black paste

Service: Heavy-duty corrosion and rust control; all metals, woods and concrete mixes; temperatures from −75 to +500F

Properties: Can be applied with brush while cold regardless of climate conditions; resists acids, salts, air, noxious fumes; water repellent; fire resistant; nonabrasive; nontoxic; acts as lubricant on interior parts

Application: Protective coating for exposed mechanisms, armatures, axles and bearings, automobile springs, underground or underwater piping systems, etc.

For more data circle MD 32, Page 229

ENGINEERING DEPARTMENT

EQUIPMENT

For additional information on this new equipment, see Page 229

Reproduction Machine 33

Charles Bruning Co. Inc., 100 *Reade St., New York* 13, *N. Y.*

Designation: BW Copyflex

Style: Diazo positive contact process reproduction; automatic

Size: 29½ w x 28 d x 50-in. high

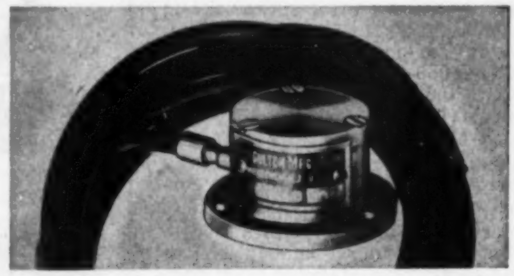
Service: Direct copy of typed, written, drawn, or line-printed material; twenty combinations of colored lines and paper plus black and white; accommodates paper to 11½-in. were at any length; 60 c 110 v a-c

Design: Automatic feed mechanism; knob on calibrated dial for speed control; self-cleaning with cup of warm water; hammertone gray finish.

For more data circle MD 33, Page 229

Accelerometer 35

Gulton Mfg. Corp., Metuchen, N. J.



Designation: Model A 403

Style: Compressional movement, piezo-electric pickup; self-generating; flange mounted

Size: Flange base 1$\frac{7}{16}$-in. diameter; unit ¾-in. high; weight 1 oz.; mounting 3—2 x 56 machine screws on 1¼-in. circle at 120-deg spacing

Service: Frequency range 3 to 4000 cps; acceleration range 0.1 to 600 g; temperature range −60 to 90 C; resonant frequency 8 kc; capacitance 1500 mmf

Design: Max. directivity perpendicular to mount; self centering suspension system to prevent breakage and slippage; units calibrated at plant at 2g from 20 to 1500 cps; 4-ft cable furnished

Application: For measuring high-frequency shock and vibration; where pulse lengths are of order of one millisecond, measure made by connecting unit into standard electronic amplifier.

For more data circle MD 35, Page 229

Accelerometer 34

Engineering Research Associates Inc., St. Paul, Minn.

Designation: Type E-62

Style: Magnetic recording, self-contained

Size: 4 x 4 x 8 in. overall; 8 lb

Service: Max recording time, 7 seconds—constant speed (±10%) recording time, 4 seconds; operating temperature 50 to 120F; seismic mass spring recording elements for following ranges of acceleration 0 to ±10, 0 to ±30, 0 to ±100, 0 to ±350 and 0 to ±500g with respective frequencies of 230, 390, 700, 1300, and 1600 cps

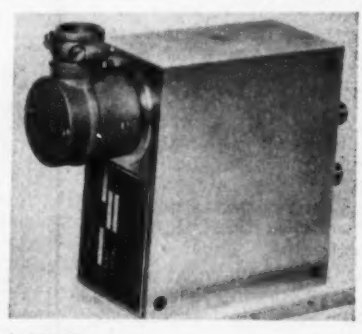
Design: Electric fuse wire starter; seismic elements exchangeable; damping of seismic elements 20% of critical; transducer and record element single unit

Application: For rocket instrumentation and operational tests of automobiles, etc., for "ride" characteristics.

For more data circle MD 34, Page 229

Differential Pressure Cell 36

Baldwin-Lima-Hamilton, Philadelphia 42, *Pa.*



Designation: FMB

Style: Opposing air bellows on cantilever; SR-4 strain gage sensing elements

Size: Pressure ranges ±10 and ±20 psi

Service: Max line pressures 50 (±10) and 100 lb (±20); input 6 and 12 v respectively, max 8 and 16 v; standard 120 and 300-ohm circuits; output at rated pressure 2.000 ±0.005mv/volt input

Design: Accuracy ¼ per cent of full scale; temperature compensated from zero to max; sensing element hermetically sealed in aluminum box; electrical connections through glass-to-metal seals; Monel bellows; mounts any position

Application: For the measurement of flow, liquid level, determination of airfoil distribution in wind tunnel and differential measure of fluid pressures.

For more data circle MD 36, Page 229

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HELPFUL LITERATURE

FOR DESIGN EXECUTIVES

78. Tube Bending Slide Chart

Carpenter Steel Co.—Slide chart presents data on desirable minimum radii for economical bending of stainless tubing and pipe. It covers tubing ranging from ⅜ to 5-in. OD in BW gages from 28 to 11. Stainless Pipe Schedules 5, 10 and 40 are described and recommended coil diameters for stainless pipe from ⅛ to 4-in. diameters are included.

79. Bellows Assemblies

Chicago Metal Hose Corp.—16-page illustrated catalog CMH-112 lists various types of Flexon bellows made of brass, stainless steel, bronze, Monel and Inconel in single and multiple ply construction. They are suitable for applications on control devices such as regulators, valves, steam traps and shaft seals as well as for expansion connections and flexible connectors for misalignment.

80. Synchronous Motors

Electric Products Co.—4-page illustrated bulletin No. 44-200 is descriptive of low-speed synchronous motors available in ratings of 20 hp and larger at speeds of 450 rpm and lower with bracket, pedestal or engine-type construction.

81. Adhesives

B. F. Goodrich Co.—8-page illustrated catalog section No. 9160 features table that lists by number each adhesive manufactured by company, materials for which it is specifically recommended and method by which most effective bond is created. Materials include buna N, neoprene, natural and GRS rubber, laminating compounds and coated fabrics.

82. Flexible Couplings

Link-Belt Co.—4-page illustrated folder No. 2363 on roller chain type flexible shaft couplings presents engineering information such as dimensions, weights, service factors and horsepower ratings. Two types of protective grease-retaining casings are described. Couplings are available with maximum bores of ¾ to 12 in.

83. Tension Control

Reeves Pulley Co.—8-page illustrated bulletin No. G-503 explains how Servo-Control slasher drive affords complete tension control at all critical points. Drive is reported to improve beam quality, increase loom output and reduce production costs through uniform slashing.

84. Synchronous Motors

Ideal Electric & Mfg. Co.—12-page illustrated brochure "Ideal Motors and Liquid Cold" describes and depicts how low-speed synchronous motors, ranging from 100 to 800 hp, drive compressors which are used to supply refrigeration to concentrators, votators in packing room and for maintaining low temperatures required in cold storage room of orange concentrate industry.

85. Fluid Motors

Denison Engineering Co.—2-page illustrated bulletin FM-3 deals with 600, 700 and 800 series constant displacement, fixed stroke fluid motors of axial piston design. Motors can be operated at 5000-psi maximum pressure at speeds from 2000 to 3000 rpm and provide maximum torque ratings of 1160, 3225 and 5585 lb-in., respectively.

86. Pipe Expansion Joint

United States Gasket Co.—Two illustrated data sheets comprise catalog No. 212 which lists uses, advantages, installation and other information on Chemiseal Teflon expansion joints for Pyrex, glass-lined and other corrosion-resistant pipe lines.

87. Power Resistor

Clarostat Mfg. Co.—Illustrated engineering bulletin 113 is descriptive of Greenohm power resistor with insulated safety knob and convenient Edison screw base. It is replaceable resistor or heater for use in tropics or where high humidity is encountered.

88. Welding Techniques

W. J. Holliday & Co.—4-page illustrated bulletin No. 907 outlines procedures to be used in welding Speed Case, Speed Treat and Speed Alloy hot-rolled alloy plate. Make and type of electrode, polarity, rod sizes, voltage and current to be used are given in table.

89. Bearing Lubrication

U. S. Electrical Motors, Inc.—4-page illustrated bulletin No. 1579 explains how Lubriflush principle enables bearings to be lubricated for life or purged of old lubricant and renewed without disturbing bearing housing.

90. Ducting Material

Arrowhead Rubber Co.—16-page illustrated catalog No. 503 lists pertinent data on Airtron Fiberglas ducting material used for flexible and rigid hose, sleeves and couplings. Information includes weights, working temperature ranges, working pressures and leakage factors. Elongation and compression limits and minimum bent radii are given for flexible convoluted and wire supported types.

91. Mechanical Tubing

Babcock & Wilcox Tube Co.—Data card No. 1150 presents information on tolerances for round seamless carbon and alloy steel mechanical tubing gives permissible variations in diameter and wall thickness. It refers to cold-drawn-unannealed and finished-annealed tubing in sizes to 8⅝-in. OD and to hot-finished tubing in sizes to 9⅝-in. OD.

92. Industrial Chain

Farrell-Cheek Steel Co.—8-page pocket-size booklet No. 40-ALG-1150 covers "85" industrial chain, chain attachments, conveyor rollers, traction wheels, sprockets and buckets.

93. Sealing Rings

Parker Appliance Co.—48-page illustrated O-ring catalog No. 903 lists dimensional and physical data for series of O-ring types including precision-molded industrial rings which meet JIC standards for air, water and hydraulic service; special formulations for use in sealing wide variety of specific liquids, gases and greases; and special grades molded to aircraft requirements.

94. Industrial Rubber Products

Barr Rubber Products Co.—4-page illustrated file-type bulletin contains information on molded, dipped, extruded and sponge rubber products which can be manufactured to user's specifications.

95. Pyrometer Accessories

Arklay S. Richards Co.—16-page illustrated catalog No. 5 describes various types and sizes of thermocouples, protection tubes, thermocouple wires, lead wires and insulators for all makes of pyrometers. Application information is included.

96. Welding Electrodes

Jessop Steel Co.—8-page illustrated booklet "Jessop Stainless Steel Welding Electrodes" contains information on selection and application of stainless steel electrodes for welding stainless steel. Current range is furnished for each type of rod in varying diameters.

FOR MORE INFORMATION

on developments in "New Parts" and "Engineering Department" sections—or if "Helpful Literature" is desired—circle corresponding numbers on either card below

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97. Precision Tubing

Tube Reducing Corp.—8-page illustrated folder R-9 on compression-formed tubing covers manufacturing process, tolerances, surface finish, mechanical properties, types, shapes and sizes. Photos show how dies compress tubing over mandrel, typical close-tolerance parts that can be made, and tapered and special internal bore shapes.

98. Disk Filters

U. S. Hoffman Machinery Corp.—2-page illustrated bulletin A-669 gives specifications and shows design of disk type filters having flow rates from 1 to 125 gpm. Units will operate at pressures up 50 psi and are easily serviced.

99. Industrial Transmissions

Drive-All Mfg. Co.—6-page illustrated bulletin lists 35 different models of industrial transmissions ranging from ½ to 30-hp capacity and designed for either vertical or horizontal shaft mounting. Units can be obtained in one, three and four-speed types with ratios varying from 1:1 to 6.4:1.

100. Rubber Products

Ohio Rubber Co.—8-page illustrated bulletin "For Direct Service on Your Rubber Needs" explains warehouse, engineering and manufacturing facilities of company. Stocked items include aprons, belting, hose, apparel, mats and other rubber products. Also shown are typical rubber parts which are produced to user's specifications.

101. Air Cylinders

Bellows Co.—24-page illustrated bulletin CL-30 deals with Controlled-Air-Power devices and tells how they were developed. Emphasis is placed upon development of precision control and air cylinder that combines cylinder, valve, valve operating controls and speed controls all in one integral unit.

102. Rubber & Plastic Calenders

Farrel-Birmingham Co.—32-page illustrated booklet describes line of calenders for processing rubber and plastics. General specifications, recent design improvements, parts lists and drawings, diagram of roll arrangements and pictures of over 30 different types and sizes of calenders are included.

103. Flexible Ball Joints

Barco Mfg. Co.—16-page illustrated catalog No. 215 "Flexible Ball Joints" describes joint which consists of short pipe nipple terminating in enlarged ball housed in outer casing between two molded gaskets. Providing 360-degree rotation and 30 to 40-degree flexing, joints are offered in 12 styles and 15 sizes from ¼ to 12 in. for pressures to 6000 psi and temperatures to 1000° F.

104. Electronic Components

Stackpole Carbon Co., Electronic Components Div.—42-page illustrated catalog RC-8 includes details of standard lines of fixed and variable resistors, line and slide switches, iron cores, choke forms and 'gimmick' capacitors. Complete mechanical and electrical specifications simplify component selection.

105. Wire Forms

E. H. Titchener & Co.—4-page illustrated folder "For Your Government Sub-Contract Work and Regular Production" contains information on design and production of wire forms, welded wire assemblies, wire and strip metal assemblies and light stampings. Reference is made to advantages of use of wire in defense production.

106. Industrial Finishing

Glidden Co. — 8-page illustrated "Glidden Industrial Review" is regular publication that contains informative articles which show production economies and sales advantages attained through selection of proper finishes for all types of products. Actual finishing problems are discussed and their solutions presented in case study form.

107. Tank Heads

Commercial Shearing & Stamping Co.—4-page illustrated bulletin "What's A-Head at Commercial" depicts and describes U-shaped, hemispherical, Obround and segmental steel tank heads developed for various industrial applications.

108. Rust Preventives

Nox-Rust Chemical Corp.—4-page illustrated bulletin discusses how Clear Coat Nos. 607 and 626 transparent rust preventives replace heavy, opague gummy compounds formerly used for export shipment as well as conventional lacquer or varnish coatings.

109. Parting Compound

Acheson Colloids Corp.—6-page illustrated bulletin No. 427 discusses uses of colloidal graphite dispersions as parting compound in metalworking, glass, rubber and corrugated boxboard industries. Unaffected by heat, material is used to prevent sticking, corrosion, galling and 'freezing' of parts.

110. Precision Stamping Facilities

John Volkert Metal Stampings, Inc.—16-page illustrated booklet "3-Way Facilities for Precision Stampings" describes company's production procedures from creat[illegible] or modifying design to engineering and m[illegible]facturing tools and dies through to produc[illegible]g in volume on automatic presses.

111. Hydraulic Equipment

Berry Motors, Inc.—8-page illustrated bulletin No. 5010 presents data on line of hydraulic pumps, motors and transmissions. Berry principle of hydraulic power is described.

112. Plastic-Fiber

Rogers Corp. — 36-page booklet "Here's Rogers and its Fiberloys" deals lightly but pointedly with wide variety of modern industrial problems, including labor unions, selling, community relations, banking and accounting. Given are facts on Fiberloy formulations of various fibrous materials. Usually in sheet form, material is used as electrical insulation, plastics, shoe materials and Duroids.

113. Aircraft Welding

Progressive Welding Sales Co.—4-page illustrated pictorial bulletin 5101 "Three-Phase Aircraft Welding of Aluminum and Alloy Steels" offers information on S3PR series of press type spot welders, S3RA series rocker arm spot welders and 3PR series combination spot and projection welders. Specifications, control features and how machines are applied to aircraft component designs are covered.

114. Corrosion Resistance Computer

H. M. Harper Co.—Operating like slide rule, pocket-sized computer accurately tells proper nonferrous or stainless steel alloys to use for 142 corrosive conditions.

115. Finned Tubing

Brown Fintube Co.—8-page illustrated bulletin No. 8 lists sizes and materials in which integral one-piece finned tubing is available. Wide variety of heating and cooling applications of this extended surface tubing is shown. Engineering data includes curves on heat transfer, friction factor and film correction. Table compares external surface of plain pipe or tubing with that of finned pipe.

116. Flat Top Conveyor Belt

Chain Belt Co.—4-page illustrated bulletin 51-59 describes flat top type of chain conveyor belt which can flex in horizontal and vertical planes as well as curve around sharp corners. Tables give strengths, weights, materials and dimensions for Rex Flextop chain, sprockets and idler disks.

117. Pipe Line Filters

Dollinger Corp.—8-page illustrated file-size bulletin gives applications and specifications for 75 filters including pressure and vacuum types. Filters feature Staynew double-action principle which provides both mechanical separation and filtration for air and other gases.

118. Relays

Potter & Brumfield—24-page illustrated catalog No. 109 contains information that will facilitate ordering procedure, aid in development problems and help select proper type relay. Overall and mounting dimensions and schematic drawings of over 150 relays for every electrical and electronic application are included.

119. Vacuum Coating Units

Distillation Products Industries—4-page illustrated data sheet details construction and performance of line of vacuum coating units. Originally used in optical field only, equipment is now used for shadowing electron microscope specimens, manufacturing telephone capacitors and reclaiming waste plastic by filming molded toys with shiny metal. Facts are summarized in chart form.

120. Industrial Brushes

Osborn Mfg. Co.—76-page illustrated catalog No. 210 lists complete line of wire and fiber wheel brushes as well as paint, varnish and maintenance brushes. Helpful information includes factors involved in selecting brush for specific job, operating equipment requirements, brush characteristics and surface speeds.

M. D. Numbers

1	21	41	61	81	101
2	22	42	62	82	102
3	23	43	63	83	103
4	24	44	64	84	104
5	25	45	65	85	105
6	26	46	66	86	106
7	27	47	67	87	107
8	28	48	68	88	108
9	29	49	69	89	109
10	30	50	70	90	110
11	31	51	71	91	111
12	32	52	72	92	112
13	33	53	73	93	113
14	34	54	74	94	114
15	35	55	75	95	115
16	36	56	76	96	116
17	37	57	77	97	117
18	38	58	78	98	118
19	39	59	79	99	119
20	40	60	80	100	120

MACHINE DESIGN 4-51

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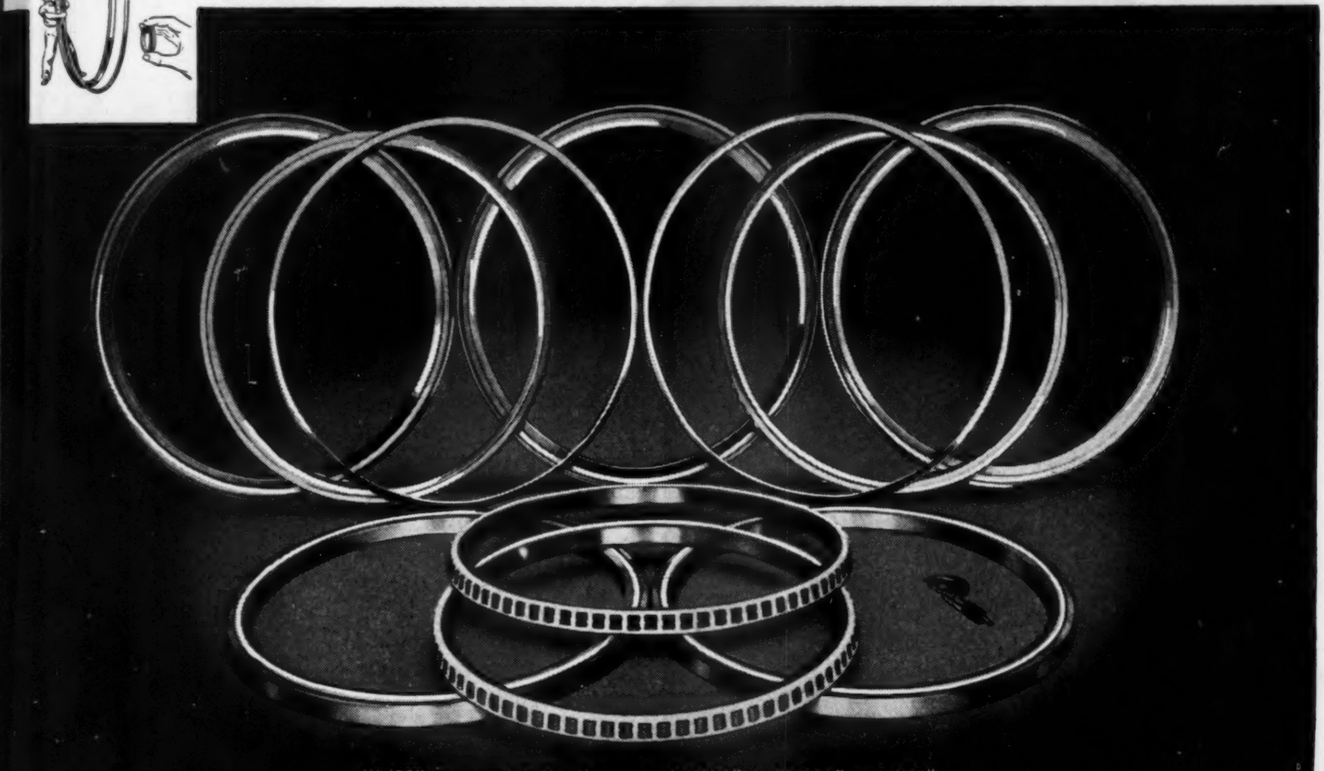

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Look again at this thin section bearing. KAYDON bearings like these, designed with very thin section, are a boon to design engineers who recognize weight-reduction and greater precision as prime problems today.

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Unique high precision techniques that hold to closest tolerances in bearings as large as 120 inches outside diameter, assure consistent accuracy in all types and sizes of KAYDON bearings and needle rollers.

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PRECISION BALL AND ROLLER BEARINGS

MEN OF MACHINES

Harold S. Sizer has been appointed director of design for machine tools of the Brown & Sharpe Co. assuming charge of engineering on machine tools and related items. A graduate of Brown University, Mr. Sizer was previously a design analyst, standards engineer and assistant director of design for the Brown & Sharpe Co.

Harold Sizer

General Electric Co. announces the appointment of **Francis K. McCune** as manager of engineering of the Apparatus Division. A graduate of the University of California, Mr. McCune joined General Electric Co. in 1928 as a student engineer on the test course. He has held positions in the International General Electric Company commercial department and in 1945 was named assistant works manager of the General Electric Co.'s West Lynn Works. In 1946 he served as a member of the Apparatu[s] Department design engineering staff and in 1948, Mr. McCune was appointed assistant to the general man- ager of the department.

Francis K. McCune

Westinghouse Electric Corp. announces the ap- pointments of the following men in the Special Prod- ucts Development Division. **Dr. S. W. Herwald**, for

merly manager of the development section, is now manager of the engineering development division; and **P. L. Fosburg,** formerly manager of the design section, is now manager of the engineering design division.

♦

Carl L. Kalitta

Carl L. Kalitta has been appointed chief design engineer of the Warner Division of Clinton Machine Co. Mr. Kalitta was previously associated with Vickers Inc. where he served in various capacities with much of his time devoted to specializing in aircraft hydraulics. He is a graduate of the Detroit Institute of Technology.

♦

R. M. Zabel has been appointed manager of engineering for the Westinghouse Electric Co.'s Lamp Division in Bloomfield, N. J. In his new capacity, Mr. Zabel will direct all the division's lamp engineering and development, its chemical and metallurgical engineering and the activity of its experimental factory.

♦

Everett S. Lee has been appointed editor of the *General Electric Review,* monthly engineering magazine published by the General Electric Co.

♦

M. J. Donovan has been appointed assistant to the president of the Franklin Supply Co.

♦

Maurice J. Erisman, chief engineer at the Los Angeles plant of Link-Belt Co., has been appointed assistant chief engineer for the company's Pershing Road Chicago plant.

♦

C. S. Beattie has been appointed executive vice president and general manager of the Delta Star Electric Co., a division of the H. K. Porter Co.

♦

Mullins Manufacturing Corp. announces the appointment of **C. L. Fix** as works manager.

♦

Richard Wheeler Jr. has joined the Mackenzie Muffler Co. as chief engineer. Prior to this, Mr. Wheeler was project engineer with Packard Motor Co.

♦

T. E. Martin, formerly assistant chief engineer with the Oliver Corp., Charles City, Iowa, has been appointed chief engineer of the company's Springfield,

Ohio plant. Mr. Martin's new position entails the direction of product design, and development and experiment testing.

♦

Alex Quayle has been appointed chief engineer in charge of all product design of the Oil Well Supply Co. a subsidiary of United States Steel Corp.

♦

Martin A. Edwards has been appointed engineering manager of the General Electric Co.'s General Engineering Laboratory. In his new capacity, Dr. Edwards will be responsible for all engineering activities of the laboratory and for engineering relations with other divisions and departments of the company.

♦

V. R. Peiffer has been appointed director of production of the Warner Division, Clinton Machine Co.

♦

General Electric Co. announces the appointment of **Henry A. Vaughn** as manager of manufacturing of the General Electric Co.'s Meter and Instrument Divisions.

♦

John Mikulak has been appointed assistant to the vice president in charge of manufacturing for the Worthington Pump & Machinery Corp. Mr. Mikulak was previously associated with the Electric Machinery Mfg. Co. and in 1944 joined the American Car & Foundry Co. where he was chief engineer of its welded products division and senior research engineer, specializing in development of welded products and methods.

♦

Milo M. Dean, former chief engineer of the Greyhound Corp., has been appointed head of the new department of development engineering of Raymond Loewy Associates.

♦

Paul Abel has been elected president in charge of engineering for the Yoder Co.

♦

Robert R. Person has been appointed staff assistant to the manager of manufacturing of the General Electric Co.'s Large Apparatus Division.

♦

John R. Howland has been appointed to head a newly created corporation office of product research for the Stewart-Warner Corp.

♦

Dr. Ralph G. Owens has been named assistant dean of engineering at Illinois Institute of Technology.

♦

Ladish Co. announces the appointments of **E. O. Dixon** as vice president in charge of research and metallurgy and **T. L. Swansen** as vice president in

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POWDER METALLURGY

PARTS MEET SEVERE REQUIREMENTS

Combining the high mechanical strength necessary to withstand heavy static, rotating or oscillating loads with a method of lubrication that maintains an unbroken oil film for thousands of hours, "COMPO" and "POWDIRON" bearings and parts offer exceptional service advantages under a wide range of operating conditions.

HIGH COMPRESSIVE STRENGTH

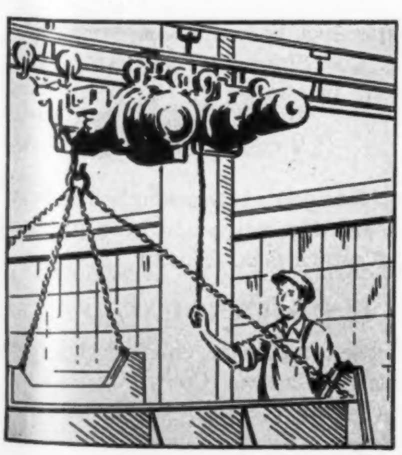
Depending on composition, "COMPO" and POWDIRON" display ultimate compressive strengths ranging from 69,000 to 140,000 psi. The porous structure is uniformly strong, since it is free from such defects as sand spots or blow holes. Permissible loads per square inch are therefore unusually high.

UNBROKEN OIL FILM

"COMPO" and "POWDIRON" bearings and parts maintain an oil film that prevents metal-to-metal contact between moving parts. When the machine is at rest, oil is stored uniformly throughout the capillary structure of the bearing or part. When the machine starts, oil is instantly fed to the surface from the microscopic pores, thus maintaining a constant oil film. For this reason, there is no theoretical top limit to the velocity of shafts operating in "COMPO" or "POWDIRON" bearings. Continuous operation of small shafts at speeds of 20,000 to 25,000 rpm has demonstrated its feasibility.

Where speed-load conditions are exceptionally severe, low-cost provisions for replenishing oil supply will keep bearings operating indefinitely.

Recommendations for specific conditions may be obtained from Bound Brook Oil-Less Bearing Company, Bound Brook, N. J.



In chain hoists and other applications where loads are heavy and maintenance difficult, the high capacity and efficient self-lubrication of "COMPO" and "POWDIRON" result in long service life with little attention.

HIGH LOAD CAPACITY

at high speeds is one of the 6* outstanding advantages of "COMPO"® and "POWDIRON"® bearings and parts

***THE 6 OUTSTANDING ADVANTAGES OF "COMPO" and "POWDIRON" are:**

1. High load capacity at high speeds
2. Extreme quietness
3. Efficient self-lubrication
4. Low installation cost
5. Low operating and maintenance cost
6. Low unit cost

Heavy *loads?* Of course — "COMPO" and "POWDIRON" bearings and parts are strong and durable. High *speeds?* With the "COMPO" and "POWDIRON" system of lubrication (oil film *constantly* maintained) there's no *theoretical* limit of velocity — *actual* shaft speeds of 25,000 rpm are not unusual.

Bearing *deliveries* are fast, too — if stock "COMPO" sizes (and there are plenty of them) meet your requirements. Write on company letterhead for latest list.

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charge of manufacturing.

Thomas Cornils has been appointed chief engineer for the Pacific Northwestern Division of the Link-Belt Co.

Robert H. Owens has been elected president and general manager of Roots-Connersville Blower Corp.

Victor F. Stine has been elected a director of the Pangborn Corp. Mr. Stine assumes the duties of vice president in charge of sales and engineering.

Howell Electric Co. announces the appointment of **Richard P. Ballou** as chief engineer. A graduate of Iowa State College, Mr. Ballou was previously chief engineer of the Feder Elecric Products Co. Prior to that, he was associated with the Raytheon Manufacturing Co., Allen-Bradley Co. and Westinghouse Research Laboratories. Mr. Ballou has been especially active in the electronics, service equipment and industrial control fields.

Emil R. Schaeffer has been appointed manager of manufacturing of the General Electric Co.'s Switchgear Divisions.

Jervis B. Webb Co. announces the appointment of **Eric O. Melmer** as chief engineer.

James M. Baker has been appointed vice president in charge of manufacturing of the Weatherhead Co. Prior to his recent appointment, Mr. Baker was factory manager at Motor Products Corp.

H. Thomas Hollowell Jr. has been elected president of the Standard Pressed Steel Co.

William A. Roberts has been elected president of Allis-Chalmers Mfg. Co. Since 1947 he had been executive vice president in charge of the tractor division.

Everett V. Allen has joined the Hughes Aircraft Co. as design engineer. Mr. Allen had been previously associated with Western Gear Works.

L. H. Hirsch has been appointed chief electrical engineer of the Century Electric Co.

Donald A. Campbell has joined Eclipse Fuel Engineering Co. as vice president in charge of engineering and research. Mr. Campbell was previously associated with Affiliated Gas Equipment Inc.

"Superior Performance"

says LOUIS ALLIS about the new VICKERS Magnetic Amplifier Control on their new *Adjustable Speed Drive*

No "warm-up" time . . . extremely fast response . . . no maintenance! These are exclusive control advantages offered by the Vickers Magnetic Amplifier in Louis Allis' Select-A-Spede—a packaged all-electric adjustable speed drive operating from A. C. circuits. Located in the Cutler-Hammer control panel of the power unit, the Vickers Magnetic Amplifier constantly regulates the field current of the generator which supplies adjustable voltage to the drive motor, providing unequalled performance advantages.

Other Vickers Magnetic Amplifier features: A-C or D-C CONTROL, A-C or D-C OUTPUT . . . RESPONDS TO SUM OR DIFFERENCE OF SEVERAL SIGNALS . . . ALLOWS ELECTRICAL ISOLATION BETWEEN CIRCUITS.

There's a VICKERS Standard Magnetic Amplifier for Your Control Needs

HIGH POWER For 60 cps power sources—27 styles—maximum output powers from 62 watts to 4200 watts.

HIGH PERFORMANCE For 60 cps power sources—28 styles—maximum output powers from milli-watts to 108 watts. For 400 cps power sources—20 styles—maximum output powers from 30 watts to 385 watts.

HIGH GAIN For 60 cps power sources—22 styles—maximum output powers from ½ watt to 1200 watts.

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THE ENGINEER'S Library

Torque Converters or Transmissions

By P. M. Heldt; published by P. M. Heldt, Nyack, N. Y.; 476 pages, 5½ by 8½ inches, clothbound; available through MACHINE DESIGN, *$6.00 postpaid.*

The complete range of torque converters and transmissions for use with combustion engines in road and rail vehicles is discussed in this book. The approach is comprehensive and concise with an adequate theoretical and practical coverage providing a useful reference for designers.

By chapter, the contents are: friction clutches; automatic clutches; hydraulic and electric couplings; overrunning clutches; principles of toothed gearing; sliding-gear and constant-mesh transmissions; bearing loads in geared transmissions; planetary transmissions; electric drive; hydrostatic transmissions; hydrokinetic torque converters; differential or power-shunt transmissions; inert type transmissions; variable-throw transmissions; automatic and power controlled stepped transmissions; pneumatic transmissions; and geared transmissions for different services. Each discussion is supported by a liberal number of diagrams, photographs and tables. Where practical, detailed studies of well-known industrial transmissions are made.

This fourth edition also includes a special 36-page appendix which describes the recent developments in passenger-car transmissions.

□ □ □

A B C of Iron and Steel

Edited by Dan Reebel, associate editor of Steel; *published by the Penton Publishing Co., Cleveland, O.; 423 pages, 8½ by 11 inches, clothbound; available through* MACHINE DESIGN, *$10.00 postpaid.*

Thirty-one authorities of the steel industry have contributed to this volume which comprehensively describes the primary processes involved in converting iron ore into finished iron and steel industrial products. The book presents the technical details of the phases of this basic industry in easily understood language.

Of particular interest to designers are the chapters that deal with the production of iron and steel shapes —pipe, wire and wire rods, forgings, gray iron castings, malleable iron castings, and steel castings. These discussions contain information relevant to design development, such as applications, costs, lim-

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to point the way to increased productivity

If you are concerned with the design or manufacture of products—military or civilian—which call for the use of metal components, you should be receiving The Alloy Pot. For 18 years this publication has had a single purpose—to point the way to increased productivity through the well-planned use of zinc die castings in product design.

As The Alloy Pot begins its 19th year, industry faces a challenge quite different from the one faced in the early 1940's. Now you are asked to build the tools of war and still maintain adequate civilian production in the face of a drain on manpower by the Armed Forces. Since zinc die castings represent the shortest distance between raw material and finished part they undoubtedly will provide the answers to many of the production problems of 1951.

The Alloy Pot will give you *specific examples* of increased productivity and lower costs through the use of zinc die castings.

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itations, advantages, properties, etc. Tables, diagrams, and illustrations are used effectively to support all discussions.

Manufacturer and Association Publications

Ball and Roller Bearing Engineering: A second edition of a text which serves as an excellent orientation to bearing engineering. Emphasis is on fundamental principles—types, forces and motions, capacity, selection and application, mounting and dismounting, lubrication and maintenance, and failures. Whenever possible, derivations and calculations involving higher mathematics have been omitted. In the section dealing with bearing types, designs used are both representative and subject to standardized manufacture; bearing applications covered are typical of representative designs. Tables of permanent reference value include: boundary dimensions for radial and thrust bearings; bearing nomenclature and capacity; general plan for boundary dimensions of ball and roller bearings; tolerances for boundary dimensions of ball and roller bearings; tolerances for boundary dimensions and running accuracy; and shaft and housing tolerances.

Authored by Arvid Palmgren, the 270-page, 7 by 10-inch, clothbound bearing manual is available from SKF Industries, Department A, Philadelphia 32, Pa., at $1.75 per copy.

Buyer's Guide for Design of Screw Machine Products: This is the second edition prepared by the National Screw Machine Products Association. The booklet includes suggestions for dimensioning drawings with respect to burring, keyway and hole location, knurling, grinding, and hardening and plating. Discussions emphasize practical engineering economy resulting from proper specification on drawings.

Copies of the 17-page, 6 by 9-inch, paper-bound booklet may be obtained from the National Screw Machine Products Association, 13210 Shaker Sq., Cleveland 20, O., at $0.25 per copy.

ASTM Standards on Copper and Copper Alloys: This latest edition brings together in convenient form the 108 American Society for Testing Materials standards pertaining to copper and copper-base alloy products. Included is a section with 20 specifications on copper, copper alloys, copper-covered steel wire, rods, bars, and stranded conductors for electrical purposes. Following is a group of 13 specifications on various nonmetals such as slab zinc, nickel, silicon copper, lead, and others. There are several sections with specifications on various copper and copper alloy products including: plate, sheet and strip; wire, rods, bars, and shapes; pipes and tubes; and die castings; and arc-welding electrodes and brazing solder. Also included in the compilation are ten test methods covering expansion, mercurous nitrate, resistivity, tension, micrographs, hardness, sampling, and grain size.

Copies of this 542-page book can be obtained from the American Society for Testing Materials, 1916 Race St., Philadelphia 3, Pa., at $4.35 each (heavy paper cover) or $5.00 (clothbound).

PROFESSIONAL Viewpoints

" . . . an interesting problem . . . but not simple"

To The Editor:

Your January, 1951 issue has an article, "Splined Positive Lock", by Thomas Barish. As you know from my article on locked gear trains, MACHINE DESIGN, December, 1948, we are interested in this subject and have done considerable study on it. Therefore, we are writing to you to point out an apparent error, since we do not know how to reach Mr. Barish.

He is quite right in stating that the situation is an *interesting* problem in the mathematics of integral numbers, but it does not yet appear *simple*. Note that his concluding formula for calculating minimum-correction tooth numbers does not always work. For example, with 23 teeth in one member, the formula works for 27 (his example), 29, and 31. It does not work for 28, 30, 32, or 33. Some other approach is needed.

As mentioned in my article, we use a tabular method, a refined sort of trial-and-error, which gives us not only minimum corrections but also corrections about the same as any error which may occur in the train. This gives us a table of corrections which allow final adjustment in one step. Mr. Barish may be interested in this scheme if his mechanisms allow for some measurement of necessary corrections.

We have abandoned our search for a direct method of making these tables since we have a hunch that there is no such method and since our approach works quickly and accurately. Therefore, you can realize our hopes and eventual disappointment at reading Mr. Barish's claim.

—R. L. BENFORD
Gear Engineering Div.
General Electric Co.
West Lynn, Mass.

" . . . data needed some extension"

To The Editor:

With reference to my article on a splined locking device in the January MACHINE DESIGN, I've found that Aeroproducts Div. of GMC have already been using the data published in this article and found it helpful. In fact, they found it needed some extension and it was my thought that you might have some way of completing it. The points that they made are covered as follows.

The final formula calls for $(1 + A)/D$ = Number of teeth to be moved on the spline with the smaller number of teeth. The spline with the larger number of teeth is to be moved this number plus 1. The Aeroproducts people pointed out that this number does not become an integer all the time. This means that it is necessary to advance the smaller spline

a certain number of teeth and the larger spline not one more, but two more or maybe three more or four more.

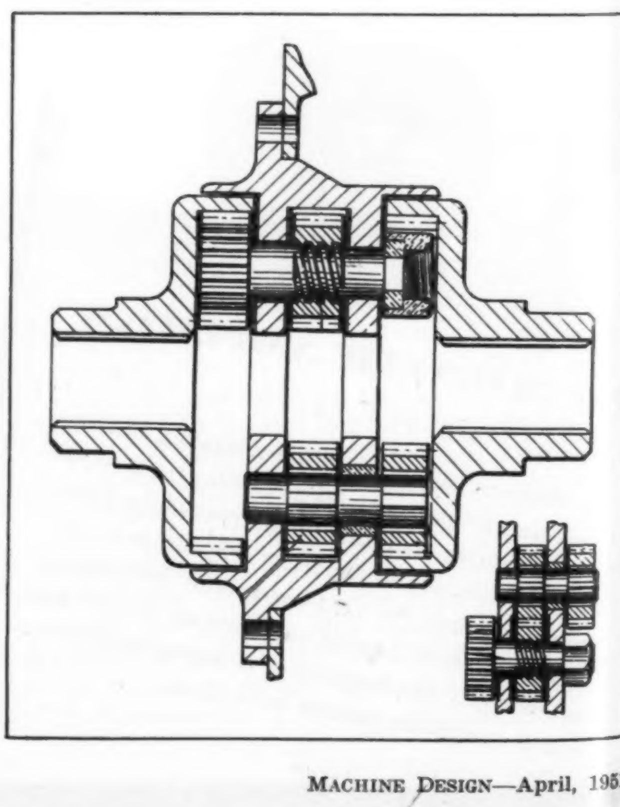
The formula for jumps of two then becomes the number of teeth to be advanced on the small spline as $(1 + 2A)/D$. In case this does not come out an integer you then try three teeth where $N = (1 + 3A)/D$.

It was gratifying to see the data prove of value so quickly. In case you do publish this as a letter to the editor or some such form, please also mention Mr. William H. Novak of Aeroproducts who worked with me on this and who discovered the continuation necessary above.

—THOMAS BARISH
Consulting Engineer
Cleveland, O.

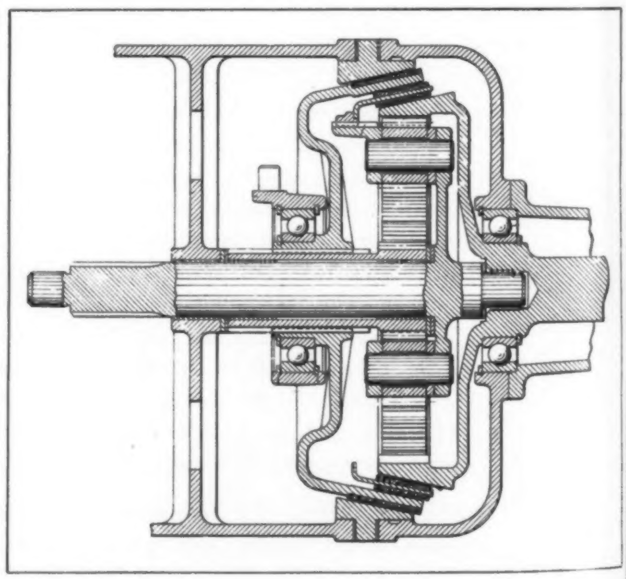
NOTEWORTHY PATENTS

DIFFERENTIAL TRANSMISSION ACTION which permits unequal torque on two drive wheels of an automotive vehicle is covered in patent 2,536,392. The device utilizes spur or helical gear trains to clamp either axle to the gear carrier when torque reaction at the other drive wheel decreases. An internal ring gear on the differential housing end of the left rear axle hub meshes with a pinion mounted on a shaft which carries a threaded section between two flanges in the gear carrier. On this threaded shaft, an in-



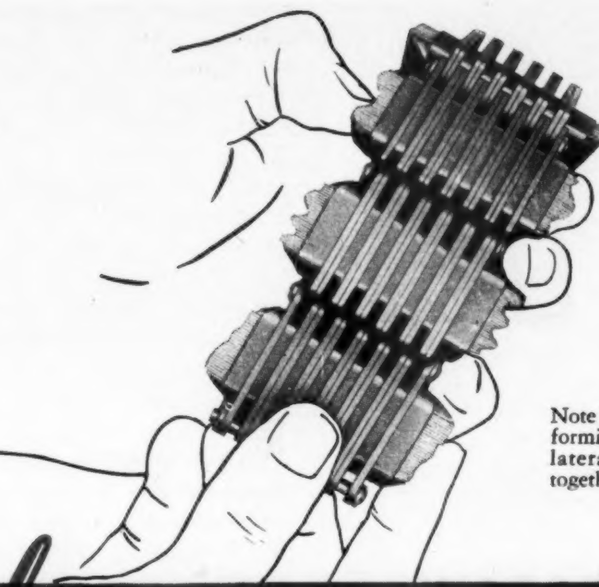
ternally-threaded pinion can move axially a short distance. The right rear axle is geared, through its ring gear and two pinions mounted on a common idler shaft, to this internally-threaded pinion. If the left wheel starts to lose traction, its ring gear moves the internally-threaded pinion axially by turning of the threaded shaft, thereby binding the pinion against one of the two gear carrier flanges. If the right wheel should lose traction, its ring gear would rotate the internally-threaded pinion through the idler gears and cause the pinion to bind on one of the flanges. The tractionless wheel is thus clamped to the driving gear carrier and the driving force is transmitted to the axle of the other wheel. Clamping action is released when torque requirements of the two wheels are equalized. In turning a corner, however, driving effort is transmitted equally to lagging and precessing wheels. Ralph R. Randall has assigned the patent to Dualoc Engineering Co.

TWO-SPEED planetary transmission is combined with a cone type automotive clutch in patent 2,510,-469 granted to Harry R. Greenlee and assigned to the Studebaker Corp. The unit employs a fork-actuated, sliding torque-transmitting member splined to a sleeve floating on the input drive shaft. This torque member has an outer friction material lined



surface and an inner surface which engages a cone-shaped braking area on the transmission housing. Shifting this member to the left locks it to the housing and holds the sun gear of the planetary gearset. Reduced speed is then imparted to the output shaft of the planetary gearing in the conventional manner.

When a direct one-to-one drive is desired, the intermediate member is shifted through the neutral position to the right, engaging its inner friction surface with an external surface of a member secured to the planet carrier. This member, in turn, engages a friction cone on the planetary ring gear to lock the



Note how slats in the self-tooth-forming chain are free to move laterally—either singly or together.

Only the P.I.V. variable speed drive uses SELF-TOOTH-FORMING CHAIN to give you positive, stepless speed changing

Link-Belt's P.I.V. is the only variable speed drive on the market today that's not dependent on friction.

That's because only Link-Belt's P.I.V. has the self-tooth-forming chain that allows *positive,* stepless speed adjustment.

If you've been looking for exact speed changing that will deliver full rated horsepower to your machines, it will pay you to call your nearest Link-Belt branch office, as our representatives are equipped to supply complete detailed information on variable speed transmission.

P.I.V. LINK-BELT
P.I.V. VARIABLE SPEED DRIVE

LINK-BELT COMPANY: Chicago 9, Indianapolis 6, Philadelphia 40, Atlanta, Houston 1, Minneapolis 5, San Francisco 24, Los Angeles 33, Seattle 4, Toronto 8, Springs (South Africa). Offices, Factory Branch Stores and Distributors in Principal Cities.

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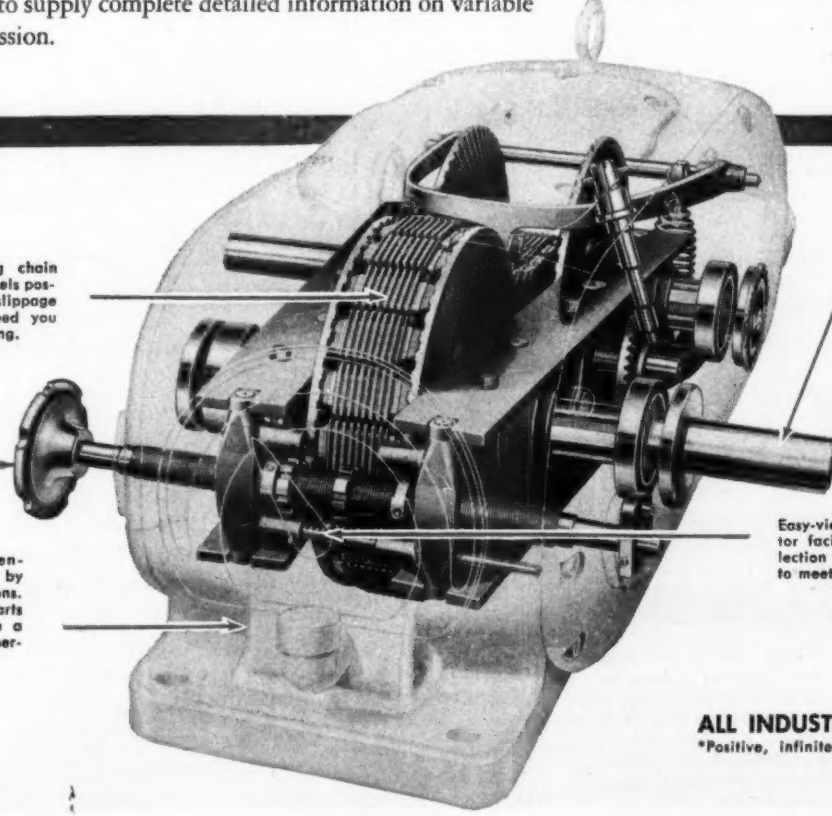
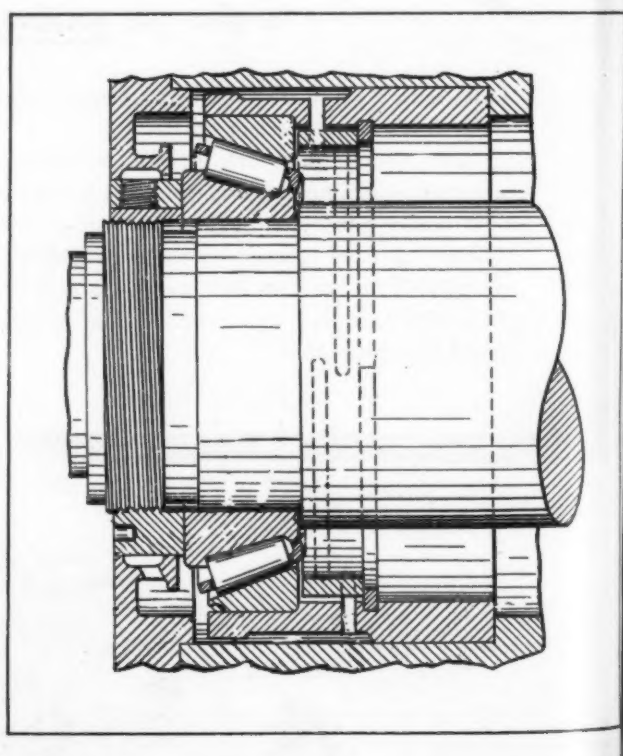


ALL INDUSTRY USES P.I.V.*
*Positive, infinitely variable

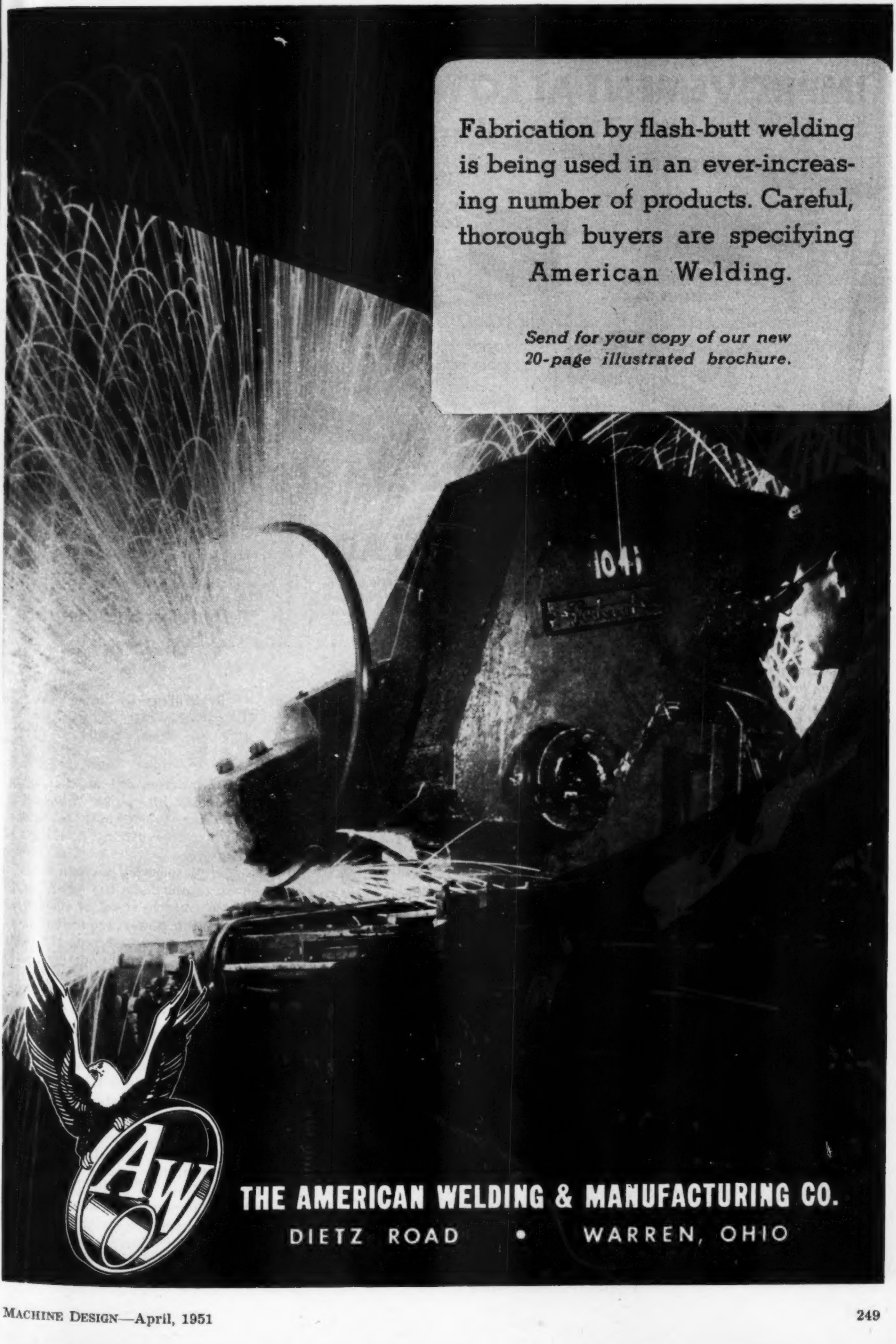
entire planet gearset together, resulting in the direct drive condition. Use of the torque-transmitting member carried by the planet carrier substantially doubles the effective clutch area, since it uses both inside and outside friction surfaces to increase the clutch capacity with no material increase in weight, size or cost. Clutch engagement can be made smoother by making the co-operating clutch surfaces of slightly different taper. One end of the cones then engages first, feathering two of the clutch members before engagement of the other member, which results in a smoother torque pickup.

AXIAL DISPLACEMENT of the outer race of the tapered roller bearing illustrated relieves the overloads caused by radial expansion. This expansion of the bearing at high temperatures created at high speeds and loads is relieved axially by a slotted sleeve in which the outer race is mounted. One end of this sleeve is secured in the bearing housing, and a series of circumferential slots in the middle section of the sleeve permits a slight axial movement of the outer race when the thrust load on the bearing reaches the



design capacity. A floating safety ring interposed between the outer race and a shoulder on the bearing shaft limits axial movement of the race and the free end of the sleeve, preventing the latter from being stressed past its elastic limit. The patent, No. 2,538,229, has been assigned to The Timken Roller Bearing Co. by Ernest G. Boden.

Complete printed copies of all patents are available from the Commissioner of Patents, Washington 25, D. C., for 25 cents each.


Fabrication by flash-butt welding is being used in an ever-increasing number of products. Careful, thorough buyers are specifying American Welding.
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AW
THE AMERICAN WELDING & MANUFACTURING CO.
DIETZ ROAD • WARREN, OHIO

Design Abstracts

(*Continued from Page* 207)

ping and threading, etc., are controlled by switch operating screws in the index drum. These functions are those of selection and not of position. The switch operating screws, therefore, need only to be taken out or put in and do not have to be adjusted. The functions of selecting the proper speeds and feeds at the proper position of the cutting tools, and of changing from rapid traverse to feed and vice versa at the proper position of the slide—functions of both selection and position—are controlled by dogs mounted in the slots on the pentagon drum, *Fig.* 3.

From a paper entitled "Application of Automatics to Small Lot Production," presented at the ASME Annual Meeting in New York, November 21-December 1, 1950. Complete copies may be obtained from ASME, 29 W. 39th St., New York 18; **$0.25** *each to members,* **$0.50** *to nonmembers.*

Dynamic Loading of Chain Drives

By William K. Stamets Jr.
Consulting Mechanical Engineer
Seattle, Wash.

CHAIN drives are an increasingly popular means for transmitting power from one shaft to another where positive driving and high efficiency are required. The present method of selecting a chain drive involves establishing the pitch on the basis of rotative speed of the sprockets and the power requirement. After the pitch is selected, the width of chain is found by determination of either an allowable chain load or an allowable horsepower per inch of width and comparison of these with design requirements. These allowable properties are based upon both pin wear and fatigue. To some extent, the effect of dynamic loading is taken into account by applying shock load factors to the design requirement and by subtracting centrifugal force directly from the allowable chain load. Often the second part of this practice leads to a chain speed limit which is unnecessarily low.

Recent experience with chain drives shows that if chain speed is increased for a given horsepower, fatigue life is likewise increased. For this rea-

son, many engineers believe that centrifugal force in the chain is passed to the sprockets and hence does not exist in the maximum-loaded portions of the chain drive.

The forces acting on the tight straight length of chain consist chiefly of a steady chain pull F_s and an inertia chain force F_c. The bottom loose length of chain is loaded mainly with the inertia force F_c.

The forces F_s and F_c in pounds can be evaluated in terms of the useful chain horsepower and velocity. Without derivation, they are

$$F_s = \frac{33{,}000\,P}{V}$$

$$F_c = \frac{qV^2}{115{,}900}$$

where V = chain speed in fpm, P = power transmitted in hp, and q = chain weight in lb per ft.

The mean force F_m and the reversed force F_r become:

$$F_m = \frac{(F_s + F_c) + F_c}{2}$$

$$= \frac{16{,}500\,P}{V} + \frac{qV^2}{115{,}900}$$

$$F_r = \frac{(F_s + F_c) - F_c}{2}$$

$$= \frac{16{,}500\,P}{V}$$

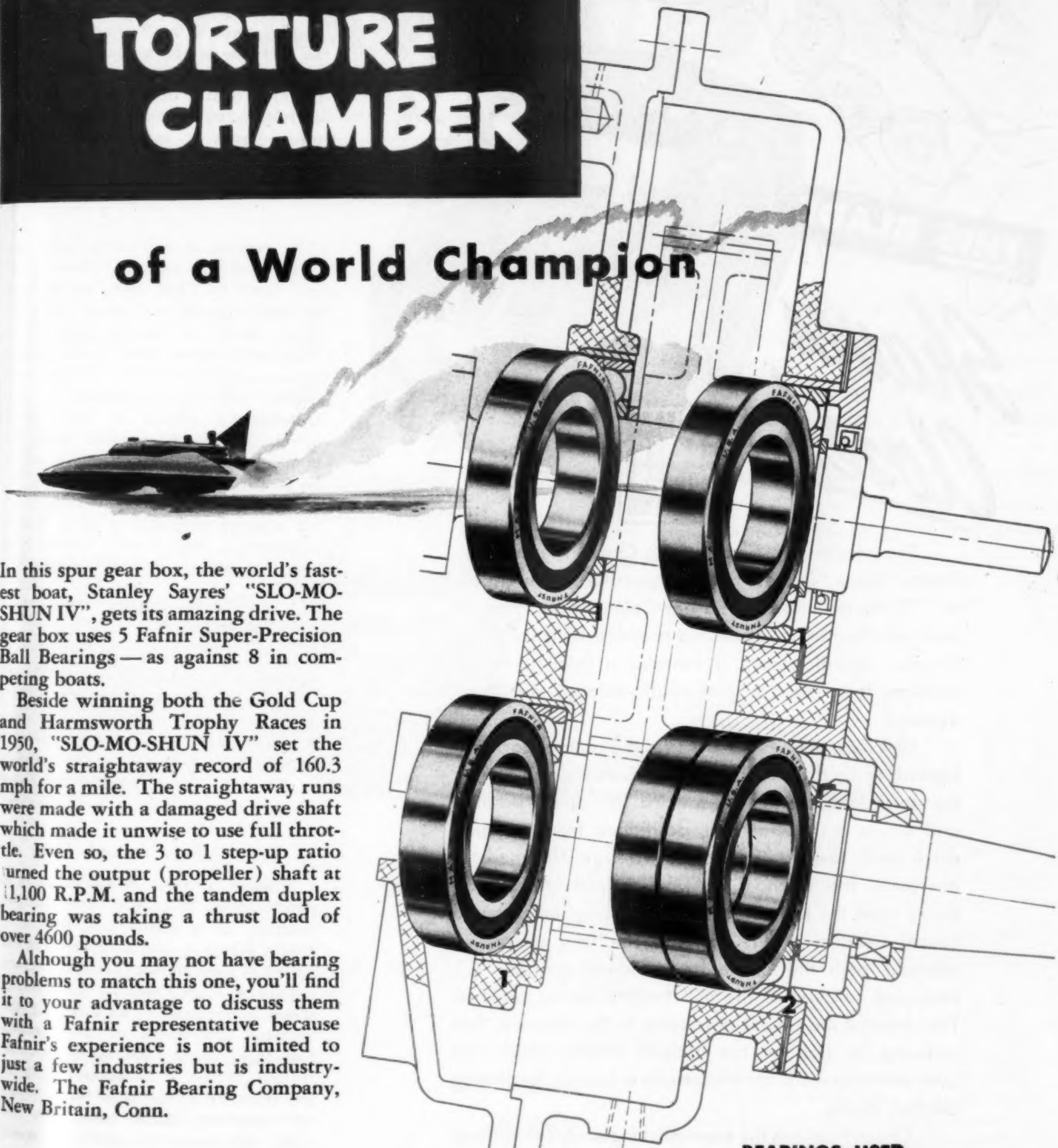
These forces are considered as fundamental loads on chain drives. Other forces, such as forces due to chordal action and sprocket impact, are considered as secondary loads. In this analysis, it is assumed that the chain sprockets have a sufficiently large number of teeth so that chordal action and sprocket impact effects are negligible.

For any given chain, the values of mean stress s_m and reversed stress s_r may be calculated using these forces. If these stresses are plotted on a Gerber failure diagram, fatigue failure may be predicted, provided that a suitable allowance is made for stress concentration.

The significance of chain speed on fatigue stresses is an important and frequently neglected subject. Assume, for example, that at a certain chain speed fatigue failure will occur because s_r and s_m are high enough to penetrate the failure boundary in the Gerber failure diagram. Then, if the chain velocity is increased 10 per cent, the steady chain pull F_s decreases 10 per cent, power transmission being held constant. The inertia force F_c under those conditions increases as the square of velocity or

BEARING TORTURE CHAMBER

of a World Champion



In this spur gear box, the world's fastest boat, Stanley Sayres' "SLO-MO-SHUN IV", gets its amazing drive. The gear box uses 5 Fafnir Super-Precision Ball Bearings — as against 8 in competing boats.

Beside winning both the Gold Cup and Harmsworth Trophy Races in 1950, "SLO-MO-SHUN IV" set the world's straightaway record of 160.3 mph for a mile. The straightaway runs were made with a damaged drive shaft which made it unwise to use full throttle. Even so, the 3 to 1 step-up ratio turned the output (propeller) shaft at 11,100 R.P.M. and the tandem duplex bearing was taking a thrust load of over 4600 pounds.

Although you may not have bearing problems to match this one, you'll find it to your advantage to discuss them with a Fafnir representative because Fafnir's experience is not limited to just a few industries but is industry-wide. The Fafnir Bearing Company, New Britain, Conn.

BEARINGS USED

1. Fafnir Super-Precision angular-contact type Ball Bearings were specified by Western Gear Works of Seattle who designed and built the gear box for "SLO-MO-SHUN IV".

2. Similar to those above except these bearings are duplexed to provide greater axial and radial rigidity.

FAFNIR

BALL BEARINGS

MOST COMPLETE LINE IN AMERICA

21 per cent. Obviously, the reversed stress s_r decreases 10 per cent. The mean stress s_m may either increase or decrease, depending upon the relative magnitudes of F_s and F_c. At low values of chain velocity, an increase in chain speed will cause s_m to decrease, while at high chain velocities, an increase in chain speed will cause s_m to increase. It can be shown mathematically that the chain speed corresponding to a minimum value of mean stress s_m is as follows:

$$V = 983 \sqrt[3]{\frac{P}{q}}$$

In general, it can be deduced that in the ordinary range of chain speed, say 1,000 to 2,500 fpm, an increase in chain velocity will cause the operating point on the Gerber failure diagram to move away from fatigue failure. Herein lies the fundamental reason that it is desirable to run transmission chains at high lineal speeds. It might appear that the upper limit of chain velocity would be that speed corresponding to $s_m = s_y$. However, this depends largely on the relative magnitudes of s_e and s_y —endurance and elastic limits— so that the optimum speed becomes:

$$V_{opt} = 983 \sqrt[3]{\frac{P\,(s_e + ks_y)}{qs_e}}$$

where k is a stress concentration factor if required. Operation of the drive at this speed corresponds to the least likelihood for fatigue failure, provided that sprockets are used having a sufficiently large number of teeth so that chordal action is ineffective.

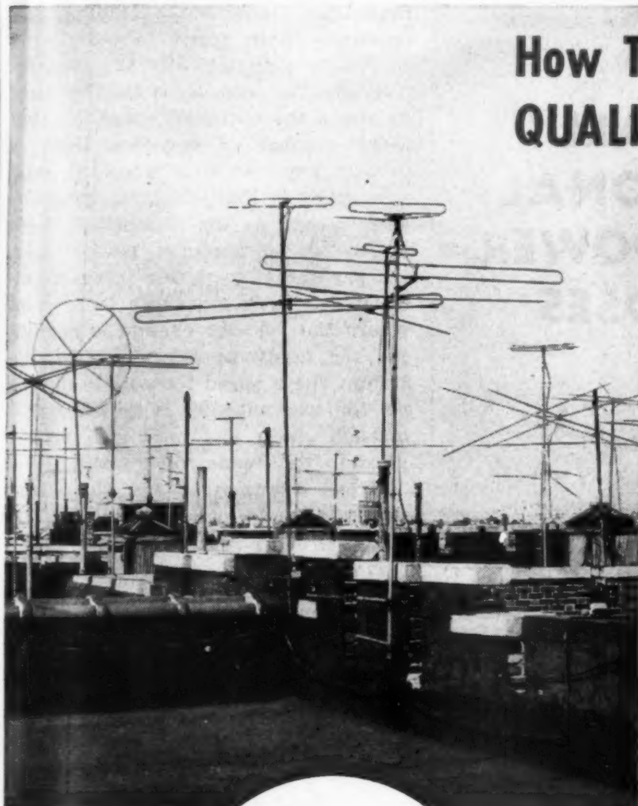
Stress Affects Variable Load

When a member is subjected to a combination of static and fatigue loading, as a chain link having a steady stress s_m and a reversed stress s_r, it has been shown that satisfactory results are obtained by considering stress concentration as affecting the variable component of the loading. For this reason, the stress concentration factor k is applied only to the reversed stress s_r in the analysis for optimum chain speed. In the final expression for optimum speed V_{opt}, the factor k is retained as a coefficient of s_y, the elastic limit. The effect of stress concentration becomes progressively less at high chain speed because, as the chain speed increases, the reversed stress s_r decreases so that the product ks_r becomes less.

Dynamic load tests prove the existence of centrifugal force on both the tight and loose lengths of chain as originally suggested. The maximum load is shown to be the sum

LAMINATES News

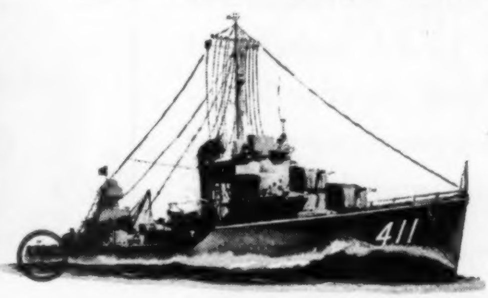
Applications for General Electric plastics laminates that can be of importance to *your* business.

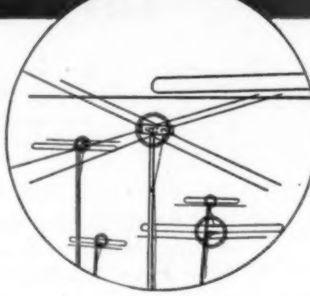


How TV benefits from the EXTRA QUALITY of G-E PLASTICS LAMINATES

Television antennas perform more efficiently when insulators are fabricated from General Electric plastics laminated sheets. G-E fine-weave, fabric-base material has excellent punching and machining properties. Its low moisture absorption makes it particularly valuable for antenna insulation.

General Electric is one of the world's largest producers and users of electrical equipment. G.E.'s years of research and experience in this field are your assurance of *extra* quality at *no* extra cost.





Bearing Down... rudder bearings and thrust plates made of G-E plastics laminates offer superior resistance to impact loads in marine applications. They have also replaced other materials in heavy-duty steel rolling mills, giving greater bearing life, reduced friction, power savings.

COMPLETE LINE OF LAMINATES

General Electric produces a complete line of plastics laminates—including sheets, tubes and rods—with cloth, paper, or special bases, or binders for a variety of applications.

G-E Plastics Speed Defense Production

NEW! Send ten cents for new G-E Laminates Comparator—a quick, easy reference to different grades, characteristics, and uses of laminated materials. Write Chemical Department, General Electric Co., Pittsfield, Massachusetts.

DEPENDABILITY THROUGH EXPERIENCE

Sheets . . . Tubes . . . Rods . . . Molded-Laminates . . . Insulating Materials

You can put your confidence in

GENERAL ELECTRIC

of the steady and centrifugal forces and the minimum chain load is the centrifugal force alone. The concept of an optimum chain speed is supported also by recent field experience with high-speed chain drives.

The optimum chain speed formula is not recommended as a "cure-all" for chain troubles. It nevertheless serves as a criterion such that when all other factors are equal, a chain running at this speed will give better service. In addition to giving more favorable fundamental loading, the optimum chain speed formula tends to reduce automatically the destructive effect of chordal action, because to attain the optimum speed, a fairly large number of sprocket teeth is usually required since sprocket rotative speed is limited. It is generally known that as the number of teeth on the small sprocket is increased, the percentage variation in chain velocity decreases markedly to a point where the velocity change is negligible for many applications. The optimum chain speed formula is intended for use with silent power-transmission chain and may also be applied to well-made roller chain.

From a paper of the same title presented at the ASME Annual Meeting in New York, November 26-December 1, 1950. Complete copies may be obtained from ASME, 29 W. 39th St., New York 18; $0.25 each to members, $0.50 to nonmembers.

Designing for Good Lubrication

By D. P. Morrell
Ford Motor Co.
Dearborn, Mich.

PREVENTIVE maintenance thinking for the most part has involved only shop practices such as scheduled repairs, lubricant handling techniques, and other functions of this nature. An equally important aspect of preventive maintenance is good maintenance features in industrial equipment design.

Good engineering practices as related to lubrication of industrial equipment require the designer to review many things which are pertinent to the good lubrication of any particular apparatus.

He should consider bearing areas and avoid loading conditions that require extra pressure lubricants.

He should consider the fact that bearings sometimes have to be replaced; therefore, provisions must

be made in the design for easy removal and replacement.

Whenever practicable, he should use a centralized system of lubrication. The reason for centralized lubrication is obvious, of course, in that it permits better shop control, less chance of points missed and considerable time saved in application of lubricants. It is also important that the designer select a system that is best for the class of equipment and type of service intended.

The designer should, whenever possible, provide complete accessibility for all lubrication components on industrial equipment. This is particularly true for pumps, reservoirs and adjustable valves.

One of the most important designer functions is to show all pertinent service information on suitable charts for use by shop personnel.

Lubrication Standards Help

The management of the Ford Motor Company some time ago recognized the need for planned lubrication practices. The management, to properly implement such a program, formed a Lubrication Equipment Sub-Committee. This Committee is made up of representatives from staff and all operating divisions of the company. As a result of the efforts of this committee and the excellent co-operation of several suppliers we have just recently publishd a set of standards entitled "Lubrication Standards for Industrial Equipment".

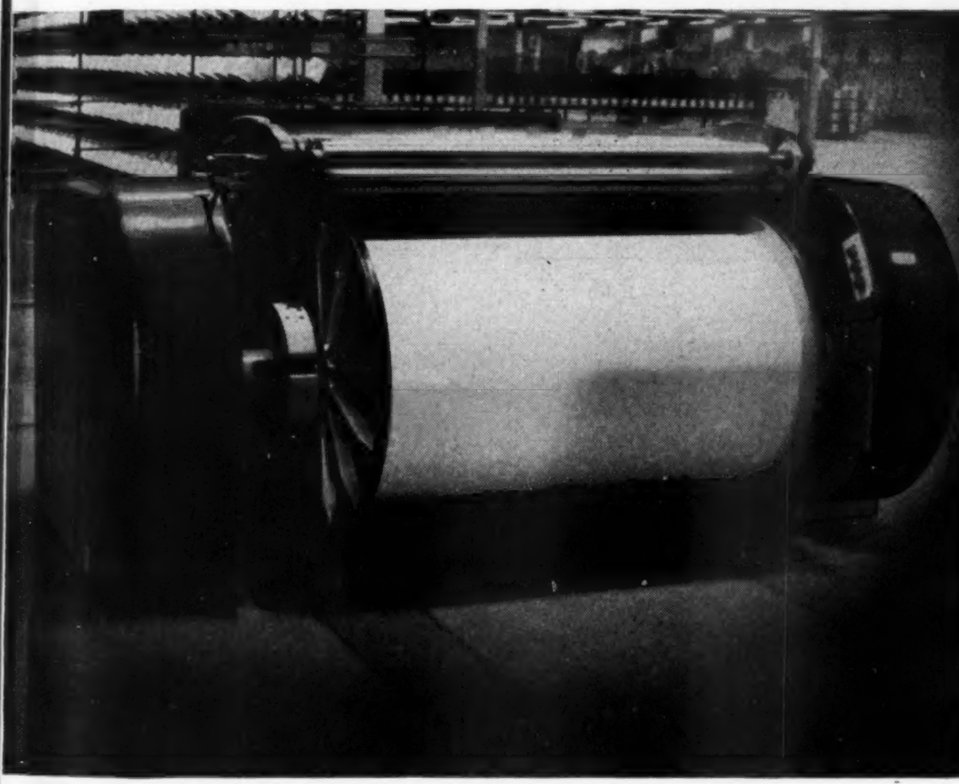
The general pattern of these standards is very similar to the JIC (Joint Industry Conference) Standards which are already in existence for electrical, hydraulic, and pneumatic applications.

These lubrication standards, by being good engineering practice standards, rather than dimensional or material standards, point out a goal in regard to performance and service accessibility, which permits the manufacturer the full scope of his know-how and ingenuity in the design of his equipment.

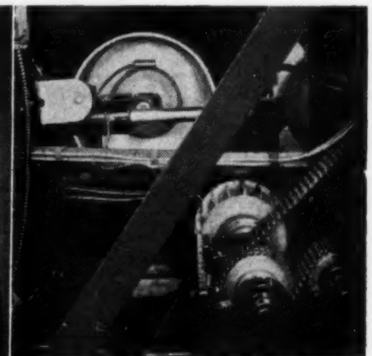
For the complete success of any company's lubrication program, it is essential that full consideration be given to both good engineering practices and good shop practices. I hope the day will soon come when the lubrication how, when, and where questions are completely and correctly answered in the design stage of industrial equipment.

From a paper entitled "Applied Lubrication in Industry," presented at the Plant Maintenance Conference in Cleveland, January 15-18, 1951.

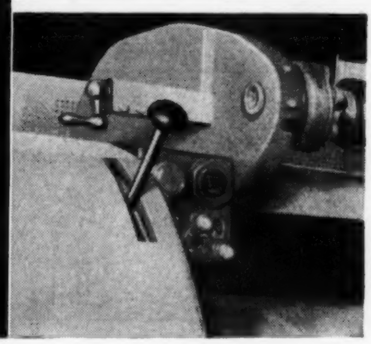
No More Lost Ends... When Warner "ICB" Units Are on the Beam!



Cocker Machine and Foundry Company's new high speed spindle driven beam warper incorporates three Warner ICB Brake Units.



Shows position of Warner 10" Brake on presser roll (upper) and 9 x 2½" Brake on main driving shaft (lower).



Shows position of Warner 5" ICB Brake Unit on crossover roll.

What Cocker says About the WARNER "ICB" Unit

Performance, cost and appearance of our high speed, spindle-driven beam warper were improved by using Warner ICB Brake Units. They provide faster stopping, thus preventing "lost ends" when a break in yarn occurs. With Warner ICB Brakes, machine stops in 1½ yds. at 500 yds. per minute machine speed. This compares with 5 yds. at the same machine speed when solenoid brakes were used. Warner ICB Units are easier to mount, require less space, eliminate noise and require no mechanical adjustments. Warner ICB Brake Power can be varied according to task, eliminating "skidding" of certain machine components. They are less expensive than the solenoid brakes formerly used.

Warner "ICB" Units are electrically-powered, low-wattage clutches and brakes. They can be operated singly or in combination — automatically or by push-button control. Being electric, they give lightning-fast response. Compact and simple, they provide easy installation and long-term dependable operation. They're a new answer to a wide variety of old machine design and operation problems. For further information on how Warner "ICB" Units may help you — write today to the Warner Electric Brake & Clutch Co., Dept. MD, Beloit, Wisconsin.

WARNER

ICB UNIT

Warner ICB Units are manufactured by Warner Electric Brake & Clutch Co.—Pioneers in the field of electric brake and clutch design and application since 1927.

News
OF MANUFACTURERS

Packard Motor Car Co. has been selected to build **General Electric Co.'s** J-47 turbojet engine for the U. S. Air Force. The contract has been quoted as one of "Detroit's biggest known defense orders to date." The company has revealed that 14 months will be required for the tooling-up job, and that 10 additional months would pass before peak production could be attained. Packard's production of the turbojet will be entirely in the company's Detroit-area plants which presently employ 9,000. The firm believes that 28,000 employees will be needed eventually for military and automobile production.

♦

Hannifin Corp., manufacturer of pneumatic and hydraulic machinery and equipment, has started construction of a new parts plant in Des Plaines, Ill. Located on a 17-acre site, the one-story factory building will cost more than $350,000 and contain approximately 40,000 sq ft.

♦

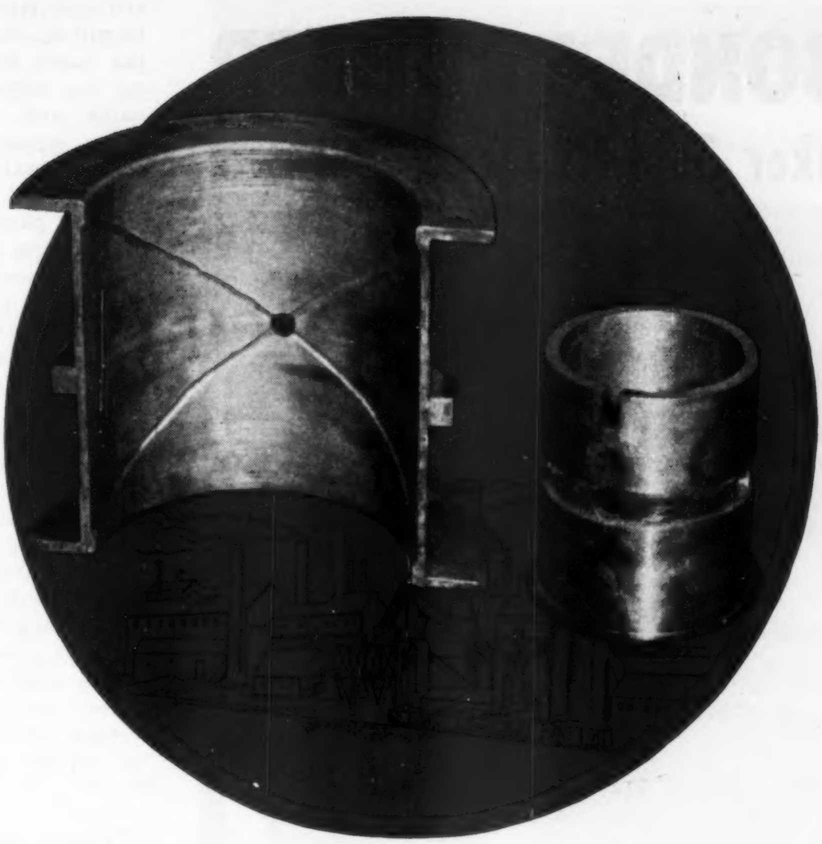
Baldwin-Lima-Hamilton Corp., Philadelphia, Pa., has been awarded a $3,000,000 contract by the U. S. Army District Engineers, Seattle, Wash. The contract calls for three 19,600-hp hydraulic turbines to be installed at the Albeni Falls Powerhouse, Priest River, Idaho.

♦

B. F. Goodrich Chemical Co.'s Port Neches, Tex., plant, used for the production of cold rubber for the government, will be increased 50 per cent. The government has authorized the installation of refrigeration equipment costing $350,000 to expand cold rubber production from 30,000 to 45,000 tons of the plant's 60,000-ton annual rated capacity.

♦

Bendix Aviation Corp. has increased employment 40 per cent since the start of the Korean war. The expansion program has affected Bendix operations in Maryland, New Jersey, New York, Iowa, Ohio, Indiana, and California: Bendix acquired a 155,000 sq ft plant at Davenport, Ia., for immediate expansion of aircraft instrument production; purchased a 175,000 sq ft plant for manufacture of field injection pumps



PRODUCED: BEARINGS THAT LAST 3 TIMES LONGER

An example showing how National Bearing Division has helped lick costly maintenance problems

AIR that's laden with dust and scale can be rough on bearings—as a large Midwest steel plant found out with an average of just 2 months' service on bearings in charging car wheels and edgers. Replacement labor and expense were going way out of line, when National Bearing Division stepped into the picture.

Our engineers made a special study of this mill's problem. As a result of this study they recommended a hydraulic bronze, both for its superior anti-frictional qualities and its high abrasion-resistance. Special methods of alloying and pouring, developed through N-B-M metallurgical research, insured sounder and denser castings of this bronze—further increasing its resistance to wear and abrasion.

Result? These bearings last 3 times as long as the bronze formerly used! Replacement labor and expense took a nosedive.

This actual example of how National Bearing Division saved money for a customer proves two important points: *first,* that N-B-M has the facilities for thorough research on proper alloying, and *second,* that we have the engineering skill to apply the fruits of this research to our customer's problems on . .

Non-Ferrous Bearings and Castings—
As-Cast or Machined

Bronze Bars—
As-Cast or Machined; Cored, Solid, Hexagon

Babbitt Metals—
For every type of bearing service

AMERICAN **Brake Shoe** COMPANY

NATIONAL BEARING DIVISION

4931 Manchester Avenue • St. Louis 10, Mo.

PLANTS IN: ST. LOUIS, MO. • MEADVILLE, PA. • NILES, OHIO • PORTSMOUTH, VA. • ST. PAUL, MINN. • CHICAGO, ILL.

and associated control equipment at Hamilton, O.; enlarged facilities at the Radio Division, Baltimore, Md., for the testing and manufacture of radar and communications equipment; expanded at the Products Division, South Bend, Ind., for accelerated work on guided missiles and aircraft parts; expanded production of autosyns and instruments at the Eclipse-Pioneer Division, Teterboro, N. J.; and enlarged facilities at the Pacific Division, North Hollywood, Calif., for development of guided missiles and telemetering apparatus.

♦

Timken-Detroit Axle Co. has launched a major expansion of its brake division manufacturing. This development is the climax of months of preparation at the division plant in Ashtabula, O. Extensive research, engineering and testing have been done on a wide variety of industrial brake applications. Typical among these are centrifuges, conveyors, cranes, winches, and other machines as well as automotive vehicles.

♦

United States Rubber Co.'s mechanical goods division has developed and started large scale production of a new line of specialties made of both plastics and rubber. The new line consists of nearly 30 types of specialties for yarn carrying operations and textile machinery. Included are such items as quills, spindle bumper tubing, lap winder rolls, spinner belts, and others. Emphasis has been placed on application of a new plastic, Uscolite, particularly suited for the textile field. Production facilities for the plastic have been doubled at the Passaic, N. J. plant.

♦

Texas Engineering & Mfg. Co. Inc. Dallas, Tex., has received a sizeable facilities contract from the Navy Department for the procurement of a full complement of machinery and equipment to be used in the manufacture of modern military aircraft. This equipment will increase production capacity to the full extent of TEMCO's building areas, which are in excess of 1,250,000 sq ft.

♦

Vanadium Corp. of America has more than doubled its production of metallurgical chrome ore as a result of the purchase by its subsidiary, Rhodesian Vanadium Corp., of properties in Southern Rhodesia, South Africa. The new properties were purchased as a going company with complete equipment, including mine

(*Continued on Page* 265)

(*Continued from Page* 262)

cars, rails, dumps, water supplies, and housing facilities for the labor force. Vanadium Corp. has also made an agreement with Societe d'Electro-Chimie of France for the use of the latter's process in the manufacture of low-carbon ferro chrome which is utilized in the production of newly developed types of alloy steels of very low carbon content. Production by this process is scheduled to begin later this year at the Niagara Falls plant at which approximately $1,500,000 is being spent to increase facilities.

•

Sessions Clock Co., Forestville, Conn., has purchased Tyniswitch Inc., Middletown, Conn., manufacturer of miniature snap-action switches. The name will be changed to Tyniswitch Division of the Sessions Clock Co. Manufacturing facilities will be located in Forestville.

•

International Rectifier Corp. has added a second story to its plant at 6908 S. Victoria Ave., Los Angeles, Calif., to meet increased production requirements of its selenium rectifiers and photocells.

•

The Duraloy Co., Scottdale, Pa., high-alloy metal producer, has announced that unfilled orders now stand at the highest level in its history, approximately $2,000,000. This represents an increase of 25 per cent over the backlog of January 1, 1951, and compares with $450,000 unfilled orders last June.

•

Standard Products Co. has established the primary policy of placing the full facilities of its seven plants at the disposal of the Armed forces, and is currently converting some of its plants to military production. The company manufactures precision metal stampings, automotive glass-run channel and weatherstripping, automotive hardware, molded and extruded rubber items, and plastics.

•

Carboloy Co. Inc., affiliate of General Electric, has launched a $2,800,000 plant expansion program to increase its production of tungsten carbides. Of the total, approximately $800,000 is being used to expand the present Carboloy plants on 8-Mile Road, Detroit, Mich., while $2,000,000 is for construction and equipment of a new plant in Edmore, Mich. At the completion of the expansion, 50 per cent of Carboloy's carbide metal out-

put will be directly for defense, under government contract, while the other 50 per cent will be indirectly for defense—in the form of carbide tools and dies needed by industry for production of defense equipment.

E. I. du Pont de Nemours & Co. is planning a major expansion of its Repauno plant, Gibbstown, N. J., and the construction of a new commercial explosives plant near Martinsburg, W. Va. Expanded facilities at Repauno will supply dimethyl terephthalate (DMT) for the company's new Fiber V plant which will be constructed at Kinston, N. C. Fiber V is a du Pont's synthetic textile fiber.

Synchro-Start Products Inc. is building a new factory in Skokie, Ill. The move was made to meet the increasing demand for automatically controlled engines and to comply with government recommendations for decentralization of critical industries.

Mack Trucks Inc., N. Y., and **Wooldridge Mfg. Co.,** Sunnyvale, Calif., have signed an agreement whereby Wooldridge will partially produce and fully assemble Mack off-highway vehicles at the Sunnyvale plant. Plans are being made for the construction of a $250,000 assembly plant.

Plax Corp., Hartford, Conn., is reactivating its specialized fabricating services on thermoplastic materials as an aid to manufacturers with military contracts involving the use of polystyrene.

Westinghouse Electric Corp. has chosen a 100-acre site four miles northwest of Elmira, N. Y., as the location for its new Electronic Tube Division. The new one-story plant, which will contain approximately eight acres of floor space, will be used to produce electronic tubes for the military services and defense industries. If military requirements drop off, the plant will be converted, rebuilt if necessary, for the manufacture of commercial tubes for radio and television transmitting and receiving, and tubes for industrial and X-ray use.

Timken Roller Bearing Co., Canton, O., is beginning a $5,503,000 expansion program to provide steels for national defense and to effect conservation of critical manufacturing materials. The expansion will involve both

the melting and fabricating facilities. By the expenditure of approximately $3,500,000 for land, buildings, three electric furnaces, additional soaking pit facilities, and related items, the Timken Steel and Tube Division will become exclusively a manufacturer of electric furnace alloy steels. A new feature of the electric furnace will be the installation of an inductive stirrer developed in Sweden recently and being used in this country for the first time.

♦

Babcock & Wilcox Tube Co. has added to its Beaver Falls, Pa., plant. Better service for users, fabricators and distributors of B&W stainless tubing will be gained by the addition.

♦

Cross Co., manufacturer of special machine tools, has added a new 10,000 sq ft plant to its facilities. The plant, which will be used for the manufacture of small machine parts, is located at 9527 Traverse Ave, Detroit, Mich. Cross has also added new equipment to its main plant at 3520 Bellevue.

♦

Joseph T. Ryerson & Son Inc., steel distributor, is expanding its St. Louis plant by reconstruction and new layout of the present plant and construction of additional warehouse space totaling approximately 50,000 sq ft. Total warehouse and office space of the enlarged plant will be about 161,000 sq ft.

♦

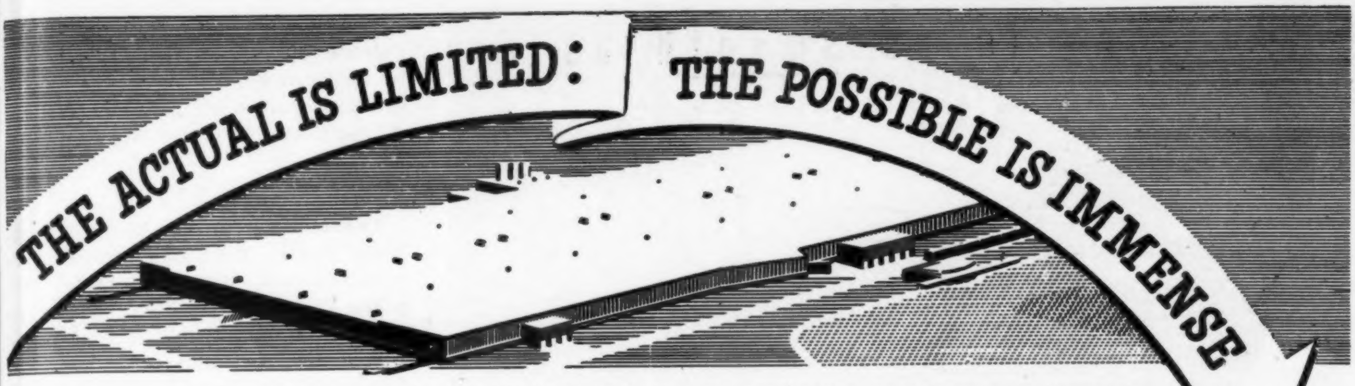
Barnett Foundry & Machine Co., Irvington, N. J., has marked its 106th year by acquiring another foundry in Dover, N. J. The new plant will provide extra capacity to meet the demands for Meehanite castings produced by Barnett.

♦

Peerless Instrument Co. has completed a 14,000 sq ft extension to research and laboratory facilities in Elmhurst, N. Y., making a total floor space of 36,000 sq ft. Peerless develops electromechanical control devices and systems, and manufactures precision parts, dies, jigs, and fixtures.

♦

Eutectic Welding Alloys Corp. is offering week-long classes for the instruction of welders in the latest metal-joining techniques. The Eutectic Welding Institute, Flushing, N. Y., is scheduled to give the course during the weeks of April 16 and May 14, 1951. Included in the training are practical applications in the joining, welding, brazing, overlaying, and cut-



NEW LINCOLN PLANT CREATED BY INCENTIVE-INSPIRED CO-ACTION IN DEVELOPING POSSIBILITIES IN PRODUCT

DESIGNS THAT UTILIZE THE ECONOMIES OF WELDED STEEL ARE ALWAYS LOWER IN COST

1 There is a 2 to 1 advantage pound per pound between the cost of steel and iron.

2 Considering both strength and safety factors, there is an additional 4 to 1 advantage with steel.

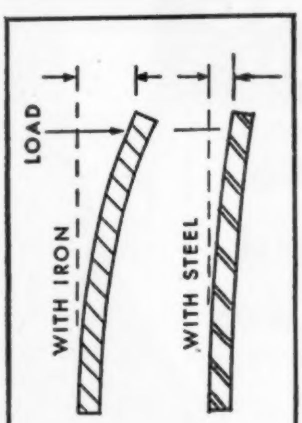
3 The design advantages are unlimited. Through origineering,* the designer is better able to locate the stronger materials in strategic areas, achieving maximum load-carrying capacity per pound of metal.

4 Production efficiency is greater. The shop can fabricate economically in any quantity . . . with a minimum of equipment and setup cost.

A Challenge to Industry

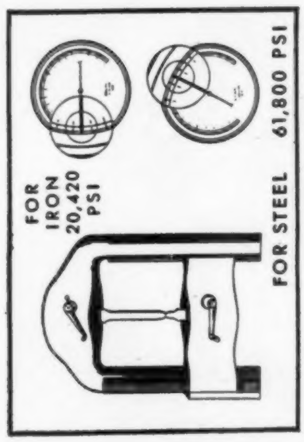
The greater potential economies possible with welded steel originate with design. To utilize the full benefits of steel requires new concepts in mechanics and appearance. Your Lincoln Welding Engineer can show you how to achieve these benefits in your designs . . . to build better products from less material at lower cost. Write The Lincoln Electric Co., Cleveland 1, Ohio.

*Origineering . . . the science of originating new engineering designs.



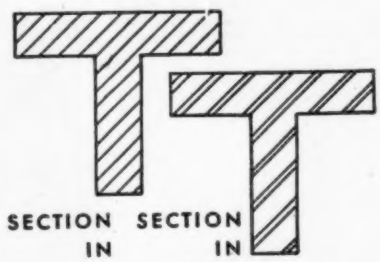
Steel is Twice as Stiff. *Only half the material is needed in welded steel designs to achieve equal rigidity.*

Steel is Ductile. *Withstands impact and shock without fracturing.*



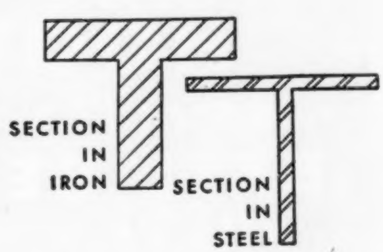
Steel is 3 Times Stronger. *Has 4 times the resistance to fatigue.*

the ACTUAL



Saves 50% Material Cost *by duplicating cross section in steel.*

increasing the YIELD



Needs 60% Less Metal *to achieve equal strength in bending.*

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ting of virtually all metals used in the defense effort. Further details may be obtained by writing to the Educational Department, Eutectic Welding Alloys Corp., 40 Worth St., New York 13, N. Y.

♦

Consolidated Vultee Aircraft Corp., Fort Worth, Tex., has increased its commitments with **Luscombe Airplane Corp.,** Dallas, Tex., to include substantial quantities of four major turret door assemblies for the B-36. Luscombe has been producing some 26 different types of door assemblies for the B-36 prior to this increase. Tools for the new assemblies have already been transferred to the Luscombe plant.

♦

Trailmobile Co. will build $2,000,000 worth of military type truck trailers for the Corps of Engineers, U. S. Army. Production will be at the company's Cincinnati, O., plant. The defense order will not interfere with civilian production of truck trailers conducted at plants in Springfield, Mo., Berkeley, Calif., and Windsor, Canada.

♦

New Britain Machine Co., has sold its Shop Furniture division to the **Industrial Bench and Equipment Mfg. Co.,** New Britain, Conn. The purchaser is housing the division in a new plant and is manufacturing and distributing under its own brand name.

♦

Alexander Brothers Belting Co., a newly formed Pennsylvania corporation, has purchased Alexander Brothers, Philadelphia, Pa., and Charlotte Leather Belting Co., Charlotte, N. C., from **Armour & Co.** The Curried Leather department was not included in the transaction and will continue to be operated by Armour. The new company will operate with Charlotte Leather as a division.

♦

Cadillac Stamp Co. is now occupying its new plant at 17315 Ryan Rd., Detroit, Mich. Cadillac manufactures marking equipment—steel stamps, machine stamps, hobs, molds, embossing dies, and hand, hydraulic and pneumatic marking machines.

♦

Aeroquip Corp., Jackson, Mich., has purchased all outstanding stock of Metalco Inc., also in Jackson. Metalco has been a substantial subcontractor of Aeroquip, furnishing a variety of special hose fittings, tube bends and elbows used in Aeroquip's hose assemblies. Operation of the newly ac-

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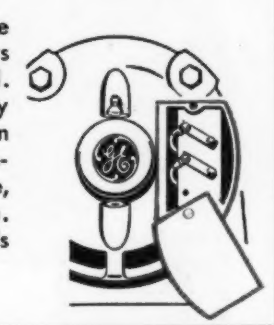
Just right for machines and compressors

More compact—20 per cent lighter weight

Capacitor is now out of the way and protected from mechanical damage by installation in the base, which is cast integrally with the frame. Conduit box is eliminated by terminal board in end shield. Weight is reduced 15 to 20 per cent depending upon the rating.

Installation made easier and quicker

Simple wiring diagram inside cover plate on end shield shows how leads are connected. Motor may be reversed by changing capacitor lead on terminal board. Cast iron construction keeps motor in line, resists rust and corrosion. Standard motor (1 to 5 hp) is dripproof.



In addition users will like the simplified transfer mechanism that provides smooth, reliable starting. The centrifugal mechanism has no rods or pins to wear; and the totally enclosed construction of the transfer switch keeps dust and dirt from the contacts. New long-life ball bearings are factory greased, but may be easily relubricated when necessary. More information on types, sizes, and characteristics in Bulletin GEA-5401. Write today. *General Electric Company, Schenectady 5, N. Y.*

**Reg. Trademark of General Electric Co.*

GENERAL ELECTRIC

754-4

quired company will be continued as a wholly owned subsidiary. Since the Metalco plant in Jackson appears inadequate to handle increased production requirements, arrangements for the lease of a larger plant in Cheboygan, Mich., have been concluded. This plant will more than triple the floor space of Metalco.

♦

E. Horton & Son Co., manufacturer of lathe chucks, is celebrating its 100th year. Flame hardened jaw ways was one of the "firsts" in the design and development of chucks introduced by Horton. In 1949 the company acquired the Gabb Mfg. Co. as a special products division producing aircraft engine timing tools and other aircraft products.

♦

Packard Motor Car Co. has received contracts amounting to $20,000,000 from the Navy Bureau of Ships for a series of powerful diesel engines and associated parts. Aircraft principles involving the use of aluminum and other lightweight materials are being incorporated into the design of these prime movers whose ratings range from 135 to 800-hp.

♦

Aluminum Co. of America, Pittsburgh, Pa., has been authorized by its stockholders to increase its indebtedness from time to time up to an aggregate total of $500,000,000 outstanding. The previous limit was $200,000,000. The increase in authorized indebtedness was recommended by the directors in connection with the company's program for construction of new plants and facilities needed to supply aluminum for national defense.

♦

Independent Pneumatic Tool Co., Aurora, Ill., manufacturer of Thor portable power tools, has purchased Armstrong-Whitworth and Co., Pneumatic Tools Ltd., Gateshead-on-Tyne, England. The Armstrong-Whitworth name will be retained.

♦

Reed Rolled Thread Die Co., Worcester, Mass., has begun construction on a new plant. Improved plant layout and added equipment will enable the company to meet the increased demand for thread rolling machines, dies, knurls, and thread rolls.

♦

Rust Engineering Co., Birmingham, Ala., is installing a $750,000 bulk-handling conveyor system to provide additional ore-handling facilities for the Alabama State Docks and Terminals, Mobile, Ala.

Society

ACTIVITIES

Announcement has been made of the cash prizes to be awarded in 1951 by the **Resistance Welder Manufacturers' Association** for the outstanding papers dealing with resistance welding subjects. The awards total $2250 and are divided as follows:

Paper emanating from an individual source, consulting engineer, private or government laboratory, or the like—$750 first prize, $500 second, and $250 third; from a University source (instructor, graduate student or research fellow)—$300 first prize, $200 second; from a University source (under-graduate students)—$250.

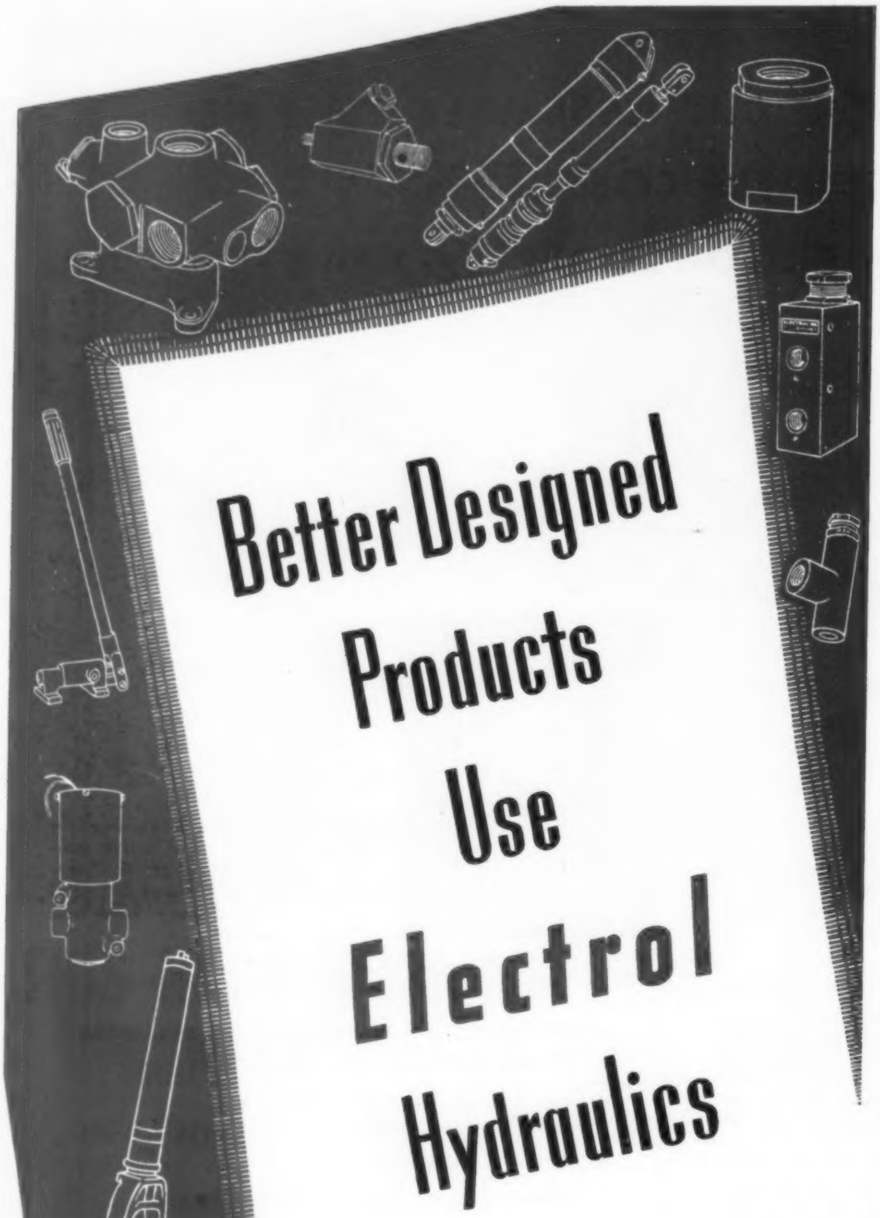
Complete contest rules may be obtained from the Resistance Welder Manufacturers' Association, 1900 Arch St., Philadelphia 3, Pa., or from the **American Welding Society,** 33 W. 39th St., New York 18, N. Y.

◆

The Defense Council of the Ingot Brass and Bronze Industry was organized on March 1, 1951, at a meeting held in Chicago of the **Brass and Bronze Ingot Manufacturers.** The Defense Council was formed to co-operate with the government agencies and to collect, collate and disseminate information with respect to and affecting the industry in connection with defense requirements. The membership is to constitute all brass and bronze ingot producers who desire to belong.

◆

In April, 1949, Doehler-Jarvis Corp., member of the **American Die Casting Institute,** donated $25,000 for the establishment of a perpetual trust to be known as the Annual Doehler Award Fund. ADCI was appointed as the trustee. Consisting of a suitable plaque and a cash honorarium of at least $500, the award is presented each September during the Annual Meeting of the Institute. The recipient is chosen by June 30 or as soon thereafter as possible. The award is based on outstanding contribution to the advancement of die casting—technical achievement, advancements in plant operations or any other activity not necessarily scientific or operational which results in the enhancement of the reputation and acceptability of die castings. Any individual, group, technical or scientific society



or committee thereof is eligible whether or not engaged in die casting business and whether or not in the employ of a member of the American Die Casting Institute. Nominations for the award and supporting papers or other material will be accepted annually from January 1 to April 30. Entries should be addressed to: Award Committee, American Die Casting Institute, 366 Madison Ave., New York 17, N. Y.

♦

During the seventh Annual Meeting of the **Society of Plastics Engineers** held in New York, Islyn Thomas, president of the Thomas Mfg. Co., Newark, N. J., was elected national president for 1951. Walter O. Bracken, Hercules Powder Co., Wilmington, Del., was elected vice president; Walter F. Oelman, vice president and general manager of Standard Molding Corp., Dayton, O., was elected secretary; and William J. Dunnican, Synvar Corp., Wilmington, Del., was elected treasurer.

♦

In a telegram to Charles E. Wilson, director of defense, the **American Society of Tool Engineers** pleged all-out support to the objective of producing for defense while maintaining essential civilian production. The ASTE, during its mobilization meeting in New York, worked on defense problems through an armed forces-industry conference, held technical sessions to exchange information on manufacturing processes, and visited defense plants to study production methods.

♦

The Franklin Institute has announced several changes among vice presidents and members of the board of managers. Edward G. Budd Jr. of the Budd Co. and James H. Robbins of the American Pulley Co. are now vice presidents. A. Felix duPont Jr., fills the vacancy on the board created by the death of his uncle, E. Paul duPont. Another new board member is James Creese, Drexel Institute of Technology.

♦

At a special meeting of the membership of the **Electric Industrial Truck Association** held in Cleveland, O., members voted to open the association's membership to manufacturers of gas-powered industrial trucks and tractors. Voted also was a change in name of the organizaiton to the **Industrial Truck Association**. The changes became effective on March 12, 1951. The association headquarters will continue to be located at 3701 N. Broad St., Philadelphia, Pa. The group of officers

elected at the annual meeting in December will serve during the remainder of 1951. These officers are: President—C. B. Cook, vice president and export manager, The Elwell Parker Electric Co., Cleveland, O.; vice president—Elmer F. Twyman, vice president, Philadelphia division of Yale and Towne Mfg. Co.; secretary-treasurer and managing director — William Van C. Brandt, Philadelphia.

•

The Board of Directors of the **Gray Iron Founders' Society** has named Donald H. Workman executive vice president. Mr. Workman has been serving as acting executive vice president since last October.

•

American Institute of Electrical Engineers has nominated officers for the 1951-52 term. Fred C. McMillan, Oregon State College, is the nominee for the presidency. Vice president nominees are: Julius C. Strasbourger, Cleveland Electric Illuminating Co.; John D. Harper, Aluminum Co. of America; Ferris W. Norris, University of Nebraska; Nelson M. Lovell, Gas, Electric Light and Power Co. of Tucson, Ariz.; and William R. Way, Shawinigan Water & Power Co., Montreal, Canada. Names of the nominees will be submitted to the more than 38,000 members of the Institute by April 15 and a report on the balloting will be made at the summer general meeting in Toronto, June 25-29.

•

New officers of the **Steel Founders' Society of America** were announced during the annual meeting in Chicago. They are: President—Thomas H. Shartle, president, Texas Electric Steel Casting Co., Houston, Tex.; vice president—G. Rhoads Casey, president and general manager, Treadwell Engineering Co., Easton, Pa.; director and executive committee member—H. A. Forsberg, vice president, Continental Foundry & Machine Co., East Chicago, Ind.

Presented during a special luncheon of the meeting were the following national medal awards:

Lorenz Memorial Gold Medal to Thomas H. Shartle, president of the Society for the past two years, and as indicated above, recently re-elected president.

Technical and Operating Medal to Paul H. Stuff, chief metallurgist, Ross-Meehan Foundries, Chattanooga, Tenn.

Steel Foundry Facts to G. A. Lillieqvist, research director and chief metallurgist, American Steel Foundries, Chicago, Ill.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DRYERS
FURNACE BLOWERS
OIL BURNERS
STOKERS
WATER PUMPS
GASOLINE PUMPS
COMPRESSORS
HOME WORK SHOPS
FANS (Ventilating)
DISHWASHERS
IRONERS
OFFICE MACHINES
FEED GRINDERS
SEPARATORS

SALES AND SERVICE

Personnel

WITH HEADQUARTERS in New York, **A. E. Whitney Jr.** has been appointed as a special representative of the chemical division of The Goodyear Tire & Rubber Co. He will offer technical services to customers in the New England and Middle Atlantic states on Goodyear's Pliovic (vinyl) resins. During the war Mr. Whitney was actively engaged in work on thin films and coated fabrics at Brooklyn Polytechnic Institute, where considerable work was done by the Quartermaster Corps. Prior to assuming his duties in New York he spent considerable time in Goodyear's own research and development laboratories, obtaining a thorough knowledge of Pliovic resins.

•

Two appointments in the industrial products sales division of The B. F. Goodrich Co. were announced recently. **A. Clarke Mack Jr.** has been named manager of flat belting, including conveyor, elevator and transmission. He succeeds **J. Robert Thompson**, who has been appointed manager of the Atlanta district for the division, where he replaces **Art Coffin**, who has retired after 35 years with the company.

•

Newly appointed manager of the central station department for the Westinghouse Electric Corp., **Carroll V. Roseberry** succeeds **R. S. Kersh**, who has been named manager of the company's steam division at South Philadelphia, Pa. Mr. Roseberry joined Westinghouse at East Pittsburgh in 1934 and two years later was assigned to the San Francisco office of the company as a central station salesman. In 1947 he was named assistant central station manager for the Pacific Coast district and in 1949 was appointed central station supervisor for the Los Angeles branch territory, from which position he comes to his new duties.

•

Edwin H. Howell of Lynnfield, Mass., has been appointed special representative of the General Electric Company's apparatus department in Washington, D. C. He formerly was manager of sales of the meter and instrument divisions, with headquarters in Lynn, Mass. In his new post,

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Mr. Howell will direct his activities specifically to liaison with governmental agencies created in connection with the national defense program and will co-ordinate his activities with those of **E. E. Potter,** General Electric vice president, who is located in Washington. **Donald E. Craig** of Marblehead, Mass., assistant manager of sales of the divisions, has been named to succeed Mr. Howell.

♦

A. William Fraser, formerly general European manager, has been appointed Midwest sales manager of Worthington Pump and Machinery Corp. He will direct the sales of the Chicago, St. Paul, Kansas City and St. Louis offices, making his headquarters in Chicago.

♦

The United States Graphite Co. of Saginaw, Mich., recently announced the appointment of **Barry Herbert Fisher** as sales representative in the Baltimore territory. Mr. Fisher will serve southeastern Pennsylvania, eastern West Virginia and North Carolina, major portions of Maryland and Delaware and all of the District of Columbia, making his headquarters in Baltimore.

♦

Herbert Gordon has been elected president of the **Sterling** Bolt Co., Chicago. This announcement was made by **Charles C. Gordon,** who revealed his resignation as president after almost one-third of a century. The resignation of **Harry Dorph,** associated with the company from its inception, was also announced. He will continue to serve on the board of directors, and **Charles C. Gordon** will become chairman of the board. **Edgar B. Miller,** also one of the guiding forces in the growth of the company, will continue as vice president and general manager, and **P. T. Phillips,** formerly secretary, has been elected vice president and secretary.

♦

To handle the sales of De Laval turbines, pumps and blowers in Wisconsin, **Patrick J. Patton** has been appointed Milwaukee manager for the commercial sales division of the De Laval Steam Turbine Co. of Trenton, N. J. He will make his headquarters at 1932 North 117th St., Wauwatosa, Wis.

♦

Robert P. Kenney, chemical sales manager of B. F. Goodrich Chemical Co., has been loaned to the National Production Authority, Washington, D. C. His new government position

has not yet been clearly defined but will entail the organization of a new unit within the plastics section of NPA to establish and direct certain plastic materials controls and distribution.

♦

Elliott Harrington has been appointed vice-chairman and secretary of a newly-established defense projects and priorities committee of General Electric's small and medium motor divisions. So that he may devote full time to this new activity, Mr. Harrington's former duties as manager of the induction motor sales division will be assumed by **R. S. Walsh,** who has been assistant manager of the division for the past year and a half.

♦

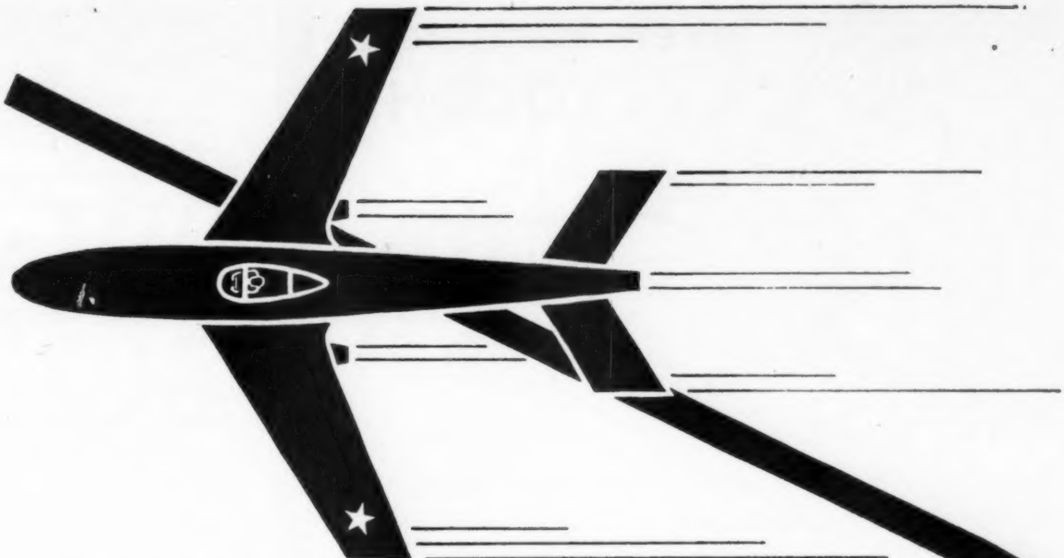
Herbert B. Nechemias was recently appointed manager of the industrial sales department at Wagner Electric Corp., succeeding **J. S. Smith,** who was appointed director of purchasing. Mr. Nechemias joined the company in 1938 as a student engineer and spent twelve years in the sales department prior to his promotion.

♦

To avoid confusion resulting from recent news stories stating that **Charles E. Wilson** is on a leave of absence from the General Electric Co., the company has issued a statement announcing that Mr. Wilson resigned his position as president last December in order to become chairman of the Defense Mobilization Board. At the time of his resignation Mr. Wilson announced that he was severing all connections with corporate and banking institutions. He was succeeded as president of General Electric Co. by **Ralph J. Cordiner,** formerly executive vice president.

♦

The board of director of the Westinghouse Electric Corp. recently elected four new vice presidents. They are **Tomlinson Fort,** who is manager of the company's headquarters apparatus sales department at Pittsburgh; **L. W. McLeod,** Southwestern district manager, with headquarters at St. Louis; **Emery W. Loomis,** Middle Atlantic district manager, Philadelphia; and **L. E. Lynde,** who has been New England district manager at Boston but now will head the company's Washington, D. C., government office. Succeeding Mr. Lynde is **E. C. Delano,** who has been a Westinghouse employee for 30 years. Upon graduation from Tufts College in 1920, Mr. Delano joined the Westinghouse graduate student training



course at East Pittsburgh, Pa., later being assigned to the industrial sales department. He was transferred to New England two years later and in 1933 was named industrial manager for the New England district, from which position he comes to his present appointment.

♦

George P. Patteson has been appointed assistant to the manager of sales of the General Electric Co. large motor and generator divisions at Schenectady, N. Y. Concurrently, **Ralph B. Bodine** was made manager and **Robert M. Easley** assistant to the manager of the large generator and converter sales division of the large motor and generator divisions.

♦

Greer Hydraulics Inc., Brooklyn, N. Y., has announced the appointment of **Harold E. Webb,** 918 North Kenilworth Ave., Glendale, Calif., as West Coast sales and service representative. Mr. Webb will handle all aviation and industrial products manufactured by Greer, including maintainance and test machines for conventional and jet aircraft and industrial hydraulic components such as accumulators, valves, filters, power units, etc.

♦

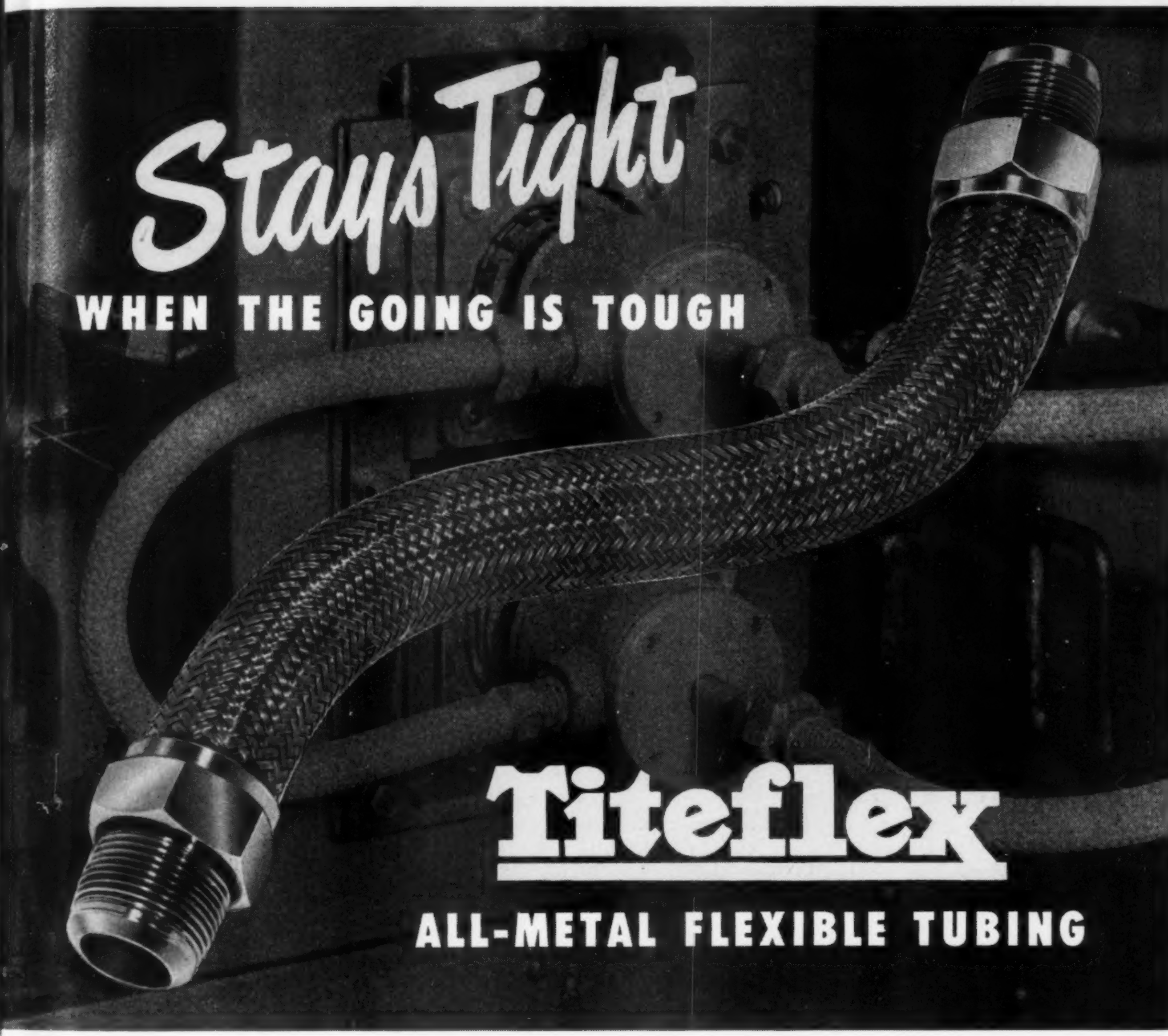
Frank R. Hunter has been named manager of a new branch office of the Allis-Chalmers Mfg. Co. general machinery division in Wichita, Kans. Mr. Hunter joined the company in 1940 and has been a sales representative in the Kansas City district office since 1942.

♦

With headquarters at the Norwood, O., works of the company, **W. A. Edwards** has been appointed district manager of the Trumbull Electric Mfg. Co. east central district. For the last three years Mr. Edwards has been switch, breaker and control sales manager at the company's main plant in Plainville, Conn.

♦

Norman W. Calkins has been named manager of tool steel sales and **Harold A. Brossman,** manager of alloy steel sales by The Carpenter Steel Co., Reading, Pa. Mr. Calkins first became associated with the company in 1946 as a service engineer in the export field. In 1949 he transferred to domestic activities in tool steel sales and until his recent promotion was assistant manager of the department. Mr. Brossman began his career with Carpenter in 1935 and during the first



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ten years developed an extensiv background on mill operations an the expediting of customer orders. H has been associated with alloy stee sales work since 1946, for the pas two years serving as assistant man ager of the department.

♦

John B. Taggart, formerly indus trial engineer for the Radio Corpora tion of America at Camden, N. J. has been appointed managing field engineer for the Work-Factor Co. management consultants, 366 Madi son Ave., New York 17, N. Y. Mr Taggart, who has been applying and instructing the work-factor system of motion time study at RCA since 1942 will specialize in service to industries in the Philadelphia area and will make his headquarters at Gwynedd, Pa.

♦

The Norge Heat division of Borg-Warner Corp. recently announced the appointment of **M. A. Straub** as sales manager of the division.

♦

John W. Nestor has been appointed assistant manager of the finishes divi sion of E. I. du Pont de Nemours and Co. Inc., succeeding the late M. A. Dibble. Simultaneously, it was announced that **Joseph B. Dietz** has become assistant director of sales, being succeeded as manager of indus trial sales by **William P. Fisher,** former assistant manager of indus trial sales.

♦

Robinson Aviation Inc., Teterboro, N. J., has announced that **H. Erich Nietsch** has joined their organization as assistant to the president. Former ly in charge of structural vibration research work at Glenn L. Martin, Mr. Nietsch will act in a liaison ca pacity between the sales and engi neering departments in the firm which manufactures airborne shock mounts and vibration isolators.

♦

Succeeding **Edward B. Wilber,** who has been elected president of Amer ican Lumber & Treating Co. at Chi cago, **Lewis P. Favorite** has been named manager of Aluminum Com pany of America's New York district sales office. Mr. Favorite, who has been serving as product manager in charge of the sale of diecastings for Alcoa, is a veteran of more than 2 years' service with the company, hav ing joined the Detroit district sales office in 1927. In 1940 he was trans ferred to the New York district sales office, remaining there until 1944, a

which time he returned to Detroit as assistant district sales manager. In 1948 Mr. Favorite was appointed St. Louis district sales manager and in October of last year was named to his most recent duties as diecastings product manager.

♦

L. J. Battaglia, newly appointed manager of the renewal sales field force of the tube department of the Radio Corp. of America, was formerly assistant to the renewal sales manager. He has been with RCA for ten years, first with the RCA Victor division in Camden, N. J., later with the RCA tube department in Harrison, N. J. In his new capacity, Mr. Battaglia will direct the activities of field representatives in the sale of tubes, parts, batteries and test equipment.

♦

R. J. Foresman, recently elected assistant secretary of the Mid-West Abrasive Co., has also been appointed assistant general sales manager. He has been employed by the firm in various capacities since 1934 and has been in the sales department for the past several years. Mr. Foresman will continue to make his headquarters at the executive offices of the company in Owosso, Mich.

♦

Appointment of **Wenzel A. Lindfors** as factory representative for the New York Belting and Packing Co. in Minnesota, northwest Wisconsin, North and South Dakota and northern Iowa, was announced recently. He will make his headquarters with W. S. Nott Co. in Minneapolis, distributors of the company's line of industrial rubber products such as conveyor and elevator belting, water, air and gasoline hose, packing, molded and extruded goods.

♦

Wyman L. Wills has been placed in charge of extruded solder sales at the Whiting, Ind. plant of Federated Metals Division, American Smelting and Refining Co. Mr. Wills comes to Whiting from Federated's St. Louis sales organization where he has been serving for the past five years.

♦

Dale D. Spoor, manager of the equipment sales department of Air Reduction Sales Co., a division of Air Reduction Co. Inc., has been granted a leave of absence to serve with the National Production Authority in Washington, D. C. Mr. Spoor, as chief of the Welding Section of the

Machinery Division of the NPA, will head the group to which manufacturers of welding equipment, apparatus, supplies, electrodes and welding rod will look for their requirements of critical materials.

♦

Willis G. Scholl has been named general sales manager of the Allis-Chalmers Mfg. Co. tractor division. He has been Eastern territory manager of the division's sales force since 1947 and in this capacity supervised the firm's industrial and farm equipment branch organization in the eastern part of the United States and Canada.

♦

P. R. Mallory Plastics Inc., a subsidiary of P. R. Mallory & Co. Inc., has announced the appointment of **Wade H. Coffing** as sales manager. In his new position, Mr. Coffing will be in charge of sales of the merchandising division in addition to the custom molding division.

♦

Robert A. Parks, Suite 827, Bowen Bldg., Washington 5, D. C., has been appointed Washington representative for the Rigidized Metals Corp., Buffalo.

♦

The appointment of two new regional sales managers for its lubricating equipment field organization has been announced by the Aro Equipment Corp., Bryan, O. **Hal Fryer,** formerly with the Davison Automobile Chemical Corp., will serve as Midwest regional sales manager, and **Harold J. Hill,** former field representative with Alemite Corp., will serve as Eastern regional sales manager.

♦

R. A. J. Wellington, general sales manager of Precision Metalsmiths Inc., Cleveland, since 1949, has been made vice president in charge of sales. Mr. Wellington was prominent in machine tool circles for 20 years when he joined the company in 1945 as Cleveland salesman.

♦

The appointment of **Donald F. Kittredge** by National Malleable and Steel Castings Co. as manager of sales, railway division, New York, was announced recently. Mr. Kittredge is replacing **Ellsworth H. Sherwood,** who was recently made assistant vice president-sales, railway division, and is being recalled to the general office of the company at Cleveland.

SALES Notes

A FIELD OFFICE in Dallas, Tex., has been opened by the **Lord Manufacturing Co.,** Erie, Pa. Located at 1613 Tower Petroleum Bldg., the new office will enable Lord field men to provide closer contact with manufacturers in the Southwest area and to give a more personal service to their problems involving vibration control and the use of bonded-rubber parts.

•

Maury E. Bettis Co., 3319 Gillham Rd., Kansas City 3, Mo., has been appointed industrial and electronic sales representative for the **Ward Leonard Electric Co.** in the Kansas City area.

•

The **Hilliard Corp.,** Elmira, N. Y., manufacturer of Hilco Lube, fuel and industrial oil filters, reclaimers and purifiers, has announced the appointment of the **Betz Engineering Sales Co.,** 1225 Magazine St., New Orleans 13, La., as a new representative. The Betz company will cover the territory of Louisiana, Mississippi and southern Arkansas, acting as agent for the full line of Hilliard-Hilco products.

•

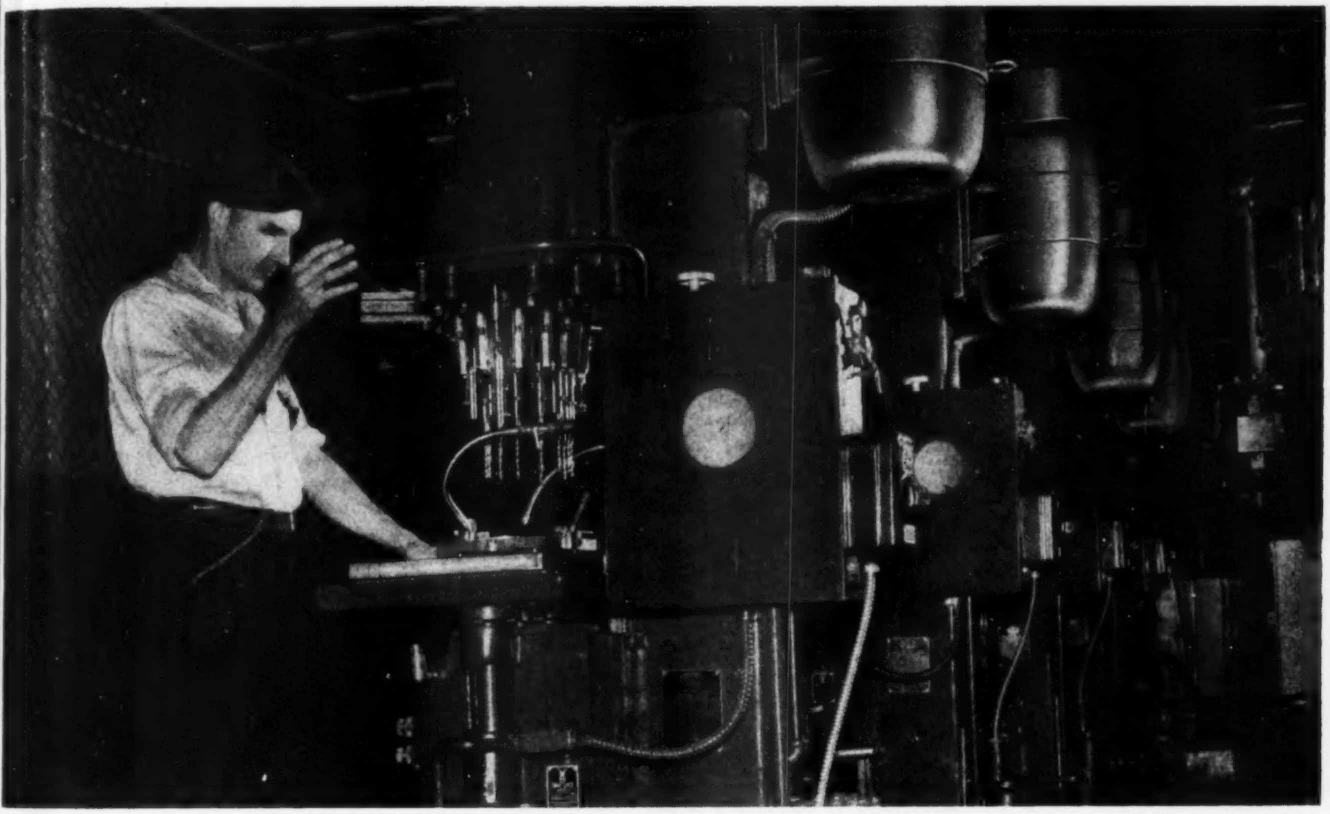
B. F. Goodrich Chemical Co. recently announced the moving of its Chicago sales office to Suite 1124, Board of Trade Bldg., West Jackson Blvd., Chicago 4, Ill. Occupancy of new quarters provides additional office and conference space required by expanding operations.

•

The San Francisco office of **Hercules Powder Co.** has been moved to the Standard Oil Bldg., 225 Bush St. The company's headquarters in this city were formerly located at 256 Montgomery St.

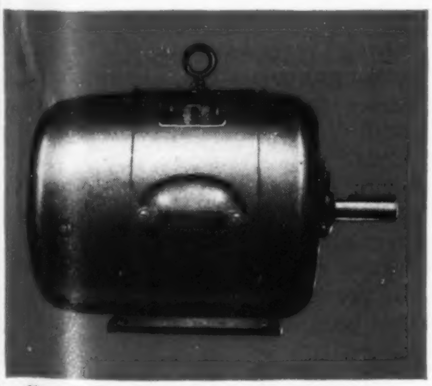
•

A. Milne & Co., tool and specialty steel distributors, New York, has announced the opening of a new tool steel warehouse at 20 South Charter St., Dayton, O., under the direction of John I. Kitts. The steels available from the company at Dayton will include hollow and solid tool steels, in standard oil hardening, air

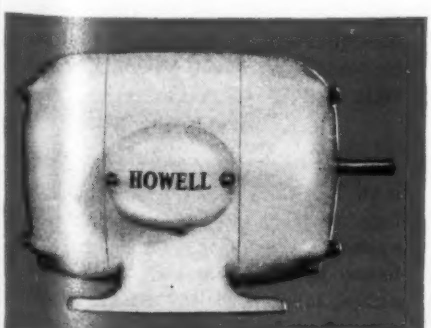


Howell totally enclosed motors power these multiple drill spindles in the plant of a leading auto maker.

HOW MUCH ARE ELECTRIC MOTORS WORTH?



Howell Type K Motor. Offers constant performance in the presence of dirt, dust, fumes and moisture. Sizes from 3 to 150 H.P. at 1800 R.P.M. Either vertical or horizontal mounting.



Howell Sanitary Motors meet the most exacting standards of the dairy and food industries. They contain no pockets, cracks, or crevices. Available for vertical or horizontal mounting.

The actual cost of the motors you see pictured here amounts to hundreds of dollars.

But their true worth, in terms of supplying the power for these multiple drill spindles to turn out finished parts, runs into thousands.

It's tough applications like this which prove the real value of Howell Industrial Type Motors. For your jobs, Howell serves three ways: (1) by engineering electric motors to your job; (2) by furnishing motors of the highest quality; (3) by serving you after the sale.

May we apply our facilities and engineering ability to your problems?

HOWELL ELECTRIC MOTORS COMPANY

Howell, Michigan

HOWELL Red Band MOTORS ®

HOWELL MOTORS

HOWELL ELECTRIC MOTORS CO., HOWELL, MICH.

Precision-built industrial motors since 1915

hardening and water hardening grades as well as hot work and shock-resisting tool steels. The tool steels stocked in Dayton will have Milne's Kolorkoted feature, the entire bar length being spray-painted an identifying color and each shipment accompanied by a heat treat card of a matching color. This feature provides permanent identification and has proved helpful in avoiding mixed stocks.

♦

Gnaedinger and Van Zelst Labquip Corp. has been appointed exclusive agent in the Chicago area for equipment made by **Tinius Olsen Testing Machine Co.,** Willow Grove, Pa. This organization, which works out of its offices at 4520 West North Ave., Chicago 39, Ill., covers the area roughly bounded by Milwaukee, Indianapolis, Fort Wayne and Davenport.

♦

The executive offices of the **Colorado Fuel and Iron Corp.** and the New York sales offices of the **Wickwire Spencer Steel Division** have been moved to 575 Madison Ave., New York 22, N. Y.

♦

A new development and manufacturing corporation specializing in valve assemblies has been established under the name **Missile Valve Co.,** 8945 Venice Blvd., Los Angeles 34, Calif. This company offers immediate service on special design and production of such items as pressure regulators, relief valves, shut-off valves or check valves. Installation of precision manufacturing facilities has been completed at the company's plant.

♦

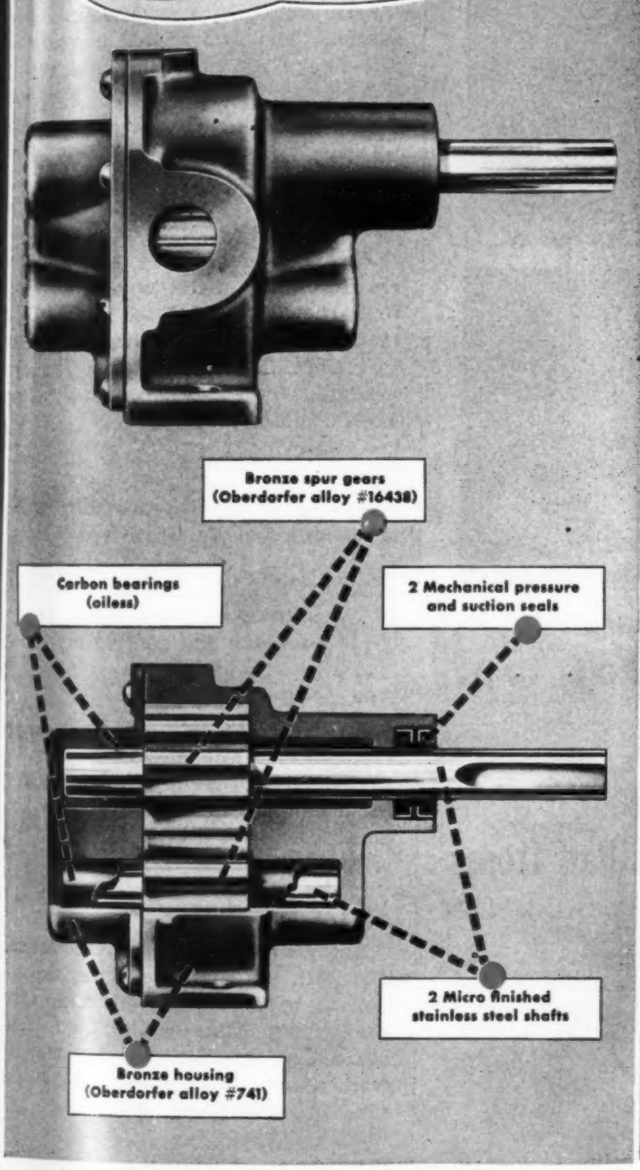
A division of the territory served by Walter G. Lee has been made by the plastics department of **Rohm & Haas Co.** William F. Condon, with headquarters in Cleveland, will serve customers in West Virginia, eastern Ohio and Western Pennsylvania.

♦

Arcos Corp., manufacturer of stainless, low-alloy high-tensile and nonferrous electrodes, Philadelphia, has opened a West Coast office and warehouse in Los Angeles. This branch office, located at 427 South Western Ave., has been opened to offer the company's customers and prospects in the southern area prompt and efficient service on its large line of welding electrodes. B. E. David has been appointed district manager.

Since the six OBERDORFER INTERNATIONAL pumps described below are new, with no prior history of use, Washington expects a DEFENSE ORDER or equivalent for their purchase. Free samples are available to O.E.M. accounts.

5 Major Points of Superiority in the New Series 1* . . .



OBERDORFER INTERNATIONAL BRONZE ROTARY GEAR PUMPS

These precision built pumps have been specifically designed for long and economical industrial service, and to reduce liquid transfer costs in the vast variety of applications where they are suitable. Their five major points of superiority incorporate all the engineering and production knowledge amassed in building literally millions of bronze pumps over a period of more than 50 years. These new Series 1 OBERDORFER INTERNATIONAL pumps will prove their worth in intensive, on-the-job tests against any competitive pumps on the market today.

CAPACITY TABLES

Gallons of Water Per Hour for Series 1 OBERDORFER INTERNATIONAL PUMPS Outlet Pressure—P.S.I. (Pounds per square inch)

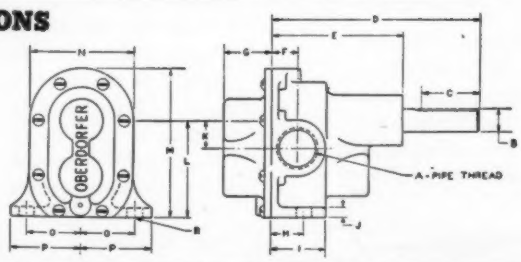
1725 RPM (Standard Electric Motor Speed)						
Pump Size	2 lbs.	20 lbs.	40 lbs.	60 lbs.	80 lbs.	100 lbs.
No. 1½	126	115	105	98	93	90
No. 2	245	225	215	205	195	190
No. 3	445	425	405	390	380	370
No. 4	640	610	580	550	530	510
No. 7	1185	1160	1140	1120	1100	1085
No. 9	1400	1375	1350	1325	1300	1280

PRICE LIST

Oberdorfer International Series 1 IMPORTANT: Order by Part Number

1½	2	3	4	7	9
Part No. 1000	Part No. 2000	Part No. 3000	Part No. 4000	Part No. 7000	Part No. 9000
$14.25	$15.75	$20.00	$22.25	$28.50	29.75

DIMENSIONS



Pump No.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	R
1½	¼	½	1	3 11/16	2 3/16	3/16	1⅛	3/16	1 1/16	3/16	15/32	1⅞	2⅞	1 15/16	⅞	1¼	5/16
2	⅜	½	1	3 11/16	2 3/16	13/32	1⅛	¾	1 3/16	¼	21/32	2 23/64	3 5/16	2⅜	1⅛	1¾	5/16
3	½	⅝	1¼	4 11/16	3	17/32	1¼	¾	1¼	¼	¾	2⅝	4	2¾	1 3/16	1⅞	13/32
4	½	⅝	1½	5 7/16	3 7/16	11/16	1¼	⅞	1 7/16	¼	¾	2⅝	4	2¾	1 3/16	1⅞	13/32
7	¾	⅝	1½	5⅜	3⅜	¾	1¼	⅞	1 5/16	5/16	1	3 5/16	5¼	3¾	1⅝	2⅛	13/32
9	1	⅝	1¾	5⅝	3⅝	⅞	1⅝	⅞	1 11/16	5/16	1	3 5/16	5¼	3¾	1⅝	2⅛	13/32

FACTS ABOUT OBERDORFER

1. The third largest non-ferrous foundry in the U. S.
2. Over a half century of bronze pump manufacturing experience
3. Long-established world-wide pump distribution
4. Millions of pumps in service
5. Capitalization over one million dollars
6. 1951 capacity for 250,000 Series 1 OBERDORFER INTERNATIONAL pumps

*First of five new series

Write Dept. MD514

INDUSTRIAL PUMP DIVISION
Oberdorfer Foundries, Inc.
Syracuse, N. Y.

Meetings

AND EXPOSITIONS

Apr. 15-18—

Scientific Apparatus Makers Association. The 33rd annual meeting to be held at the Greenbrier Hotel, White Sulphur Springs, W. Va. Kenneth Andersen, 20 North Wacker Drive, Chicago 6, Ill., is executive vice president.

Apr. 16—

Packaging Machinery Manufacturers Institute. Semi-annual meeting to be held at the Hotel Dennis, Atlantic City, N. J. Additional information may be obtained from society headquarters, 342 Madison Ave., New York 17, N. Y.

April 16-18—

Society of Automotive Engineers. Aeronautic and aircraft engine display meeting to be held at the Statler Hotel, New York. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Apr. 16-18—

American Society of Lubrication Engineers. National convention to be held at the Bellevue-Stratford Hotel, Philadelphia, Pa. W. F. Leonard, 343 South Dearborn St., Chicago, 4, Ill., is secretary.

Apr. 17-20—

American Management Association. The 20th national packaging exposition to be held in the Atlantic City Auditorium, Atlantic City, N. J. Additional information may be obtained from society headquarters, 330 West 42nd St., New York 18, N. Y.

Apr. 18-21—

National Screw Machine Products Association. The 18th annual meeting to be held at Netherland Plaza Hotel, Cincinnati, O. Additional information may be obtained from National Screw Machine Products Assoc., 13210 Shaker Square, Cleveland 20, O.

Apr. 22-26—

American Ceramic Society. The 53rd annual meeting to be held at the Palmer House, Chicago, Ill. Additional information may be obtained from American Ceramic Society, Columbus, O.

Apr. 25-26—

Metal Powder Association. Seventh

annual meeting to be held at Hotel Cleveland, Cleveland, Ohio. Additional information may be obtained from society headquarters, 420 Lexington Ave., New York 17, N. Y.

Apr. 28-May 14—

Paris Fair held at the Paris Fair Exhibition Grounds, Paris, France. Additional information may be obtained from Rowland Associates, 420 Lexington Ave., New York 17, N. Y.

Apr. 30-May 4—

Materials Handling Exposition to be held in the International Amphitheatre, Chicago, Ill. Additional information may be obtained from Clapp and Poliak, Inc., 341 Madison Ave., New York 17, N. Y.

May 16-18—

Society for Experimental Stress Analysis. Annual spring meeting to be held at the Wardman Park Hotel, Washington, D. C. Additional information may be obtained from Dr. Edward Wenk Jr., c/o The David Taylor Model Basin, Washington 7, D. C.

May 23-24—

American Society for Quality Control. Fifth annual convention to be held at the Hotel Cleveland, Cleveland, Ohio. Additional information may be obtained from society headquarters, 22 East 40th St., New York 16, N. Y.

May 24-25—

Society of the Plastic Industry. Annual national meeting to be held at the Greenbrier Hotel, White Sulphur Springs, W. Va. W. T. Cruse, 295 Madison Ave., New York 17, N. Y., is executive vice president.

May 28-June 8—

Canadian International Trade Fair to be held at the Exhibition Grounds, Toronto, Canada. Additional information may be obtained from R. G. Pendrith, Publicity Section, Canadian International Trade Fair, Toronto, Canada.

June 11-16—

National Congress of Applied Mechanics to be held at Illinois Institute of Technology, Chicago, Ill., under the sponsorship of the ASME, ASCE, AIChE, AMS, APS, IAS, SESA, U. S. National Committee on Theoretical and Applied Mechanics, Illinois Tech, Purdue University, Northwestern University and University of Illinois. Lloyd H. Donnell, Illinois Institute of Technology, 3300 South Federal St., Chicago 16, Ill., is general chairman.

New Machines

Business Equipment

OFFICE TYPEWRITER: Series 6 model with new Page Gage for indicating end of paper. Two graduated rings on platen are set for size paper being used. Red signal shows when 2½ in. of paper remain; scale gives remaining paper down to ½-in. of end. Adjustable for paper lengths from 7 to 14 in. Machine includes deepened segment which gives typebar increasing support as it swings up, enlarged platen. Printing surface slanted at 12½ degrees from vertical for maximum legibility. Finished in gray with green keyboard. *L. C. Smith & Corona Typewriters Inc., Syracuse, N. Y.*

Domestic

REFRIGERATOR: Imperial two-door, 10-cu ft model with three different storage areas. Top freezing compartment holds 73 lb frozen foods, refrigerated by coils on five sides of compartment. Shelf area cooled by Refrig-O-Plate on back wall, automatically defrosts without attention, includes 18.5 sq ft shelf area. Hydrators have transparent tops, can be stacked on top of each other, are cooled by coils in cabinet walls. All refrigerated areas cooled by same compressor unit. *Frigidaire Division, GMC, Dayton, O.*

ELECTRIC RANGE: RO-60 and RO-50 models have one large oven that converts into two separate ovens by using a movable partition. Upper oven has two heating units for broiling, baking or roasting. Lower oven has single bottom heating unit for baking or roasting. Both ovens have separate controls. Moving center dividing section to bottom provides single 6160 cu in. oven. Ranges have short interval and 1½-hr timers, clock control for appliances. *Frigidaire Division, GMC, Dayton, O.*

Heating and Ventilating

INDUSTRIAL HUMIDIFIER: Uses no ducts, pumps, compressed air or steam lines. Requires only water feed line and 110-v, 60-cycle current. Vapor emitted in fine mist form. Unit works on principle of atomization through centrifugal action. Vaporization capacity, one gal per hour; weight, 50 lb; dimensions, 12 by 14 by 22 in. Includes

¼-hp motor. *Industrial Materials Purchasers, New York, N. Y.*

HEATERS: Direct-fired units for applications including indirect circulating heating using heat transfer mediums such as Dowtherm, and for direct heating of vapors and liquids. Capacities, 100,000 to 15,000,000 Btu per hour. Complete automatic control systems available. *Struthers Wells Corp., Warren, Pa.*

DEHUMIDIFIER: Adsorption type package unit gives precise humidity control in temperature range from minus 40 to plus 100 F. Can maintain humidity levels as low as 15 per cent in enclosed areas up to 25,000 cu ft. Three-channel continuous adsorption unit has plug-in automatic operation. Dimensions, 45 by 19 by 16 inches. Runs on 110-v, 60-cycle, single-phase current with max power consumption of 1.2 kw. *Dryomatic Corp., Alexandria, Va.*

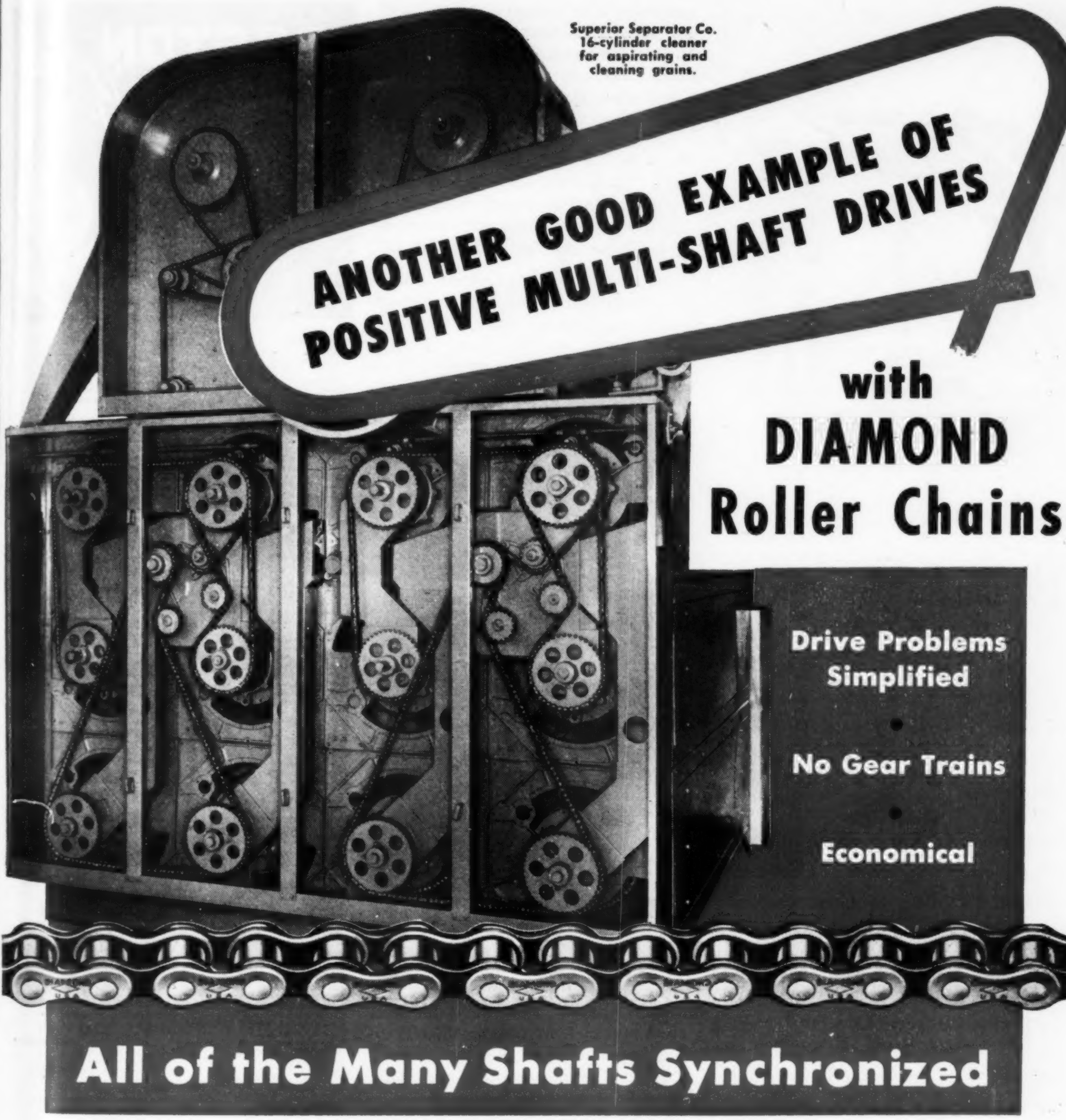
ROOM VENTILATOR: Twin-fan unit with automatic controls. Air circulates until temperature drops to selected degree, then shuts off. If room temperature rises, fans automatically turn on. Portable, or will fit double-hung or casement windows at least 17-in. wide. Fans swivel to draw in air from outside or blow air out window. Three-speed fans deliver 3450 cu ft of air per minute at distance of three feet from fan when at top speed. *General Electric Co., Bridgeport, Conn.*

PORTABLE HUMIDIFIER: Electric Moisture Magnet unit for home or commercial use. Removes 14 pints of water every 24 hours from room air at 80 F and 79 per cent relative humidity or up to 1 pint every hour under more severe conditions. Fan draws damp air over refrigerated coils, expels dry reheated air. Unit measures 30½ in. high by 13 in. wide by 13 in. deep; weight, 56 lb. *Remington Air Conditioning Div., Remington Corp., Auburn, N. Y.*

Heat Treating Equipment

ELECTRIC FURNACE: Model FG-7800 furnace for heavy-duty heat treating work. Provides temperatures to 2700 F. Can be supplied gastight with atmosphere connections. Separate control panel equipped with voltage-regulating muti-tap transformer, temperature indicator and controller, magnetic contactor, and high limit cutoff. Overall size, 33¼ by 39¼ by 64 in.; loading area, 18 by 24 by 18 in. *Pereny Equipment Co., Columbus, O.*

HEAT TREATING UNIT: Semiautomatic, controlled-atmosphere model T-100-E for bright heat treating, carburizing, carbonitriding, annealing, and copper brazing. Rated at 100 lb



• It would be difficult to conceive of any more economical and practical means of driving the many shafts at the speeds and directions of rotation required than is accomplished by the Diamond Roller Chains shown on the grain cleaner above.

The dependable long-life performance of Diamond Chains keeps machinery output up. They do not slip and operate equally well over or under sprockets. For drive applications of all kinds, the data in new Catalog 709 will help you.

DIAMOND CHAIN COMPANY, Inc., Dept. 435, 402 Kentucky Ave., Indianapolis 7, Indiana.

Offices and Distributors in all principal cities

Write for new 40-page Catalog 709

DIAMOND ROLLER CHAINS

per hour, with temperatures to 2100 F. Hearth measures 12 by 18 by 10 in. Unit consists of electrically heated furnace sealed to combination cooling chamber and quenching tank. Cooling chamber water jacketed, with automatic temperature controls. Quench tank has automatic built-in oil heating and cooling coils. All doors and quenching elevator are pneumatically operated. *Ipsen Industries Inc., Rockford, Ill.*

BAKING OVEN: Small gas-fired industrial oven for baking or tempering small production orders, for heating samples, and for shop or laboratory research or testing. Outside dimensions, 14 by 16 by 20 in.; heating chamber, 10 by 9 by 12 in., with two shelves. Capacity, 250 to 650 F. Includes pyrometer actuated controller, dial thermometer, two 60-minute timers. Weight 75 lb. *The Carlson Co., New York, N. Y.*

Maintenance

WORK TOWER: Electric-hydraulic platform designed for one-man operation. Moto-Lift has capacity of 400 lb, weighs 600 lb. Powered by 6-v battery through hydraulic pump. Standard battery raises tower 45 times without recharging. Platform raises from 7 to 17 ft in 25 seconds, lowers at rate of one ft per second. Safety tube prevents rapid fall of platform in event of hydraulic failure. Base folds to 40 by 65 in. to permit passage through standard 7-ft high doors. Safety brake on each caster locks wheels when working. *Safway Steel Products Inc., Milwaukee, Wis.*

Manufacturing

SHEAR: Capacity, 20 ft of ½-in. mild steel plate. Unit weighs 135,000 lb, has speed of 20 strokes per minute. Equipped with hydraulic holddowns capable of exerting holding force of 70 tons. Holddown pistons equipped with automotive type piston rings. Squaring shear has 24-in. throat and 48-in. back gage range. *The Cincinnati Shaper Co., Cincinnati, O.*

PUNCH PRESS: Deep throat, 4-ton capacity model. Open height with ram up increased to 8 in.; press punches to center of 18-in. circle. Frame construction strengthened and knock-out added. *Benchmaster Manufacturing Co., Los Angeles, Calif.*

BAND SAW: Contour-matic machine equipped with 36-in. stroke, heavy-duty hydraulic work table for precision machining of heat-resisting alloy components in jet engine production. Table handles up to 1 ton, has feed rate variable to 18 fpm

WHY DOES A BOSTON *Gear* *Sprocket*...

PUT EXTRA MONEY IN YOUR *Pocket?*

"Sprockets are sprockets", you may say. *"Any old sprocket will do. Might as well pick 'em up cheap."*

But stop and think! Roller chain costs money—is **increasingly hard to get.** Cheaply made sprockets with carelessly cut teeth chew up chain **fast** — reduce its life 20% or more. Sprockets that do not run true — that lack balance and correct tooth formation — waste power, too.

Chain—power—cost of shutdowns for replacement — check these against the small "saving" in first cost between a "bargain counter" sprocket and a sprocket of genuine BOSTON *Gear* quality design, materials, workmanship, finish and accuracy. **BOSTON Sprockets are cut with the same care and precision as BOSTON Gears.**

With Sprockets as with Gears, Reductors, Ratiomotors, Bearings and hundreds of other items of transmission equipment, it pays to standardize on BOSTON — readily available from nearby stock. (See adjacent list of Authorized Boston Gear Distributors).

BOSTON *Gear* Sprockets and Chain cover a complete range of chain drive service from "flea" power to forty horse power.

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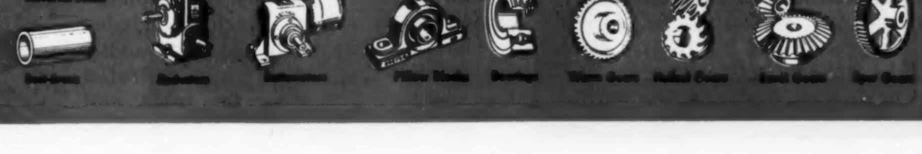


with rapid return. Table carried on rollers, driven by 200-psi hydraulic pressure. Both feed pressure and rate of travel can be controlled. Cutting speeds infinitely variable between 40 and 10,000 fpm. *DoAll Co., Des Plaines, Ill.*

VERTICAL MILLING ATTACHMENT: Converts horizontal millers for precision vertical milling. Drive is through spiral bevel gears, with spindle runout held within 0.00015-in. full indicator reading. Vertical spindle speed 1/3 higher than horizontal spindle speed. Designed for 1 hp at 2000 rpm. *W. H. Nichols Co., Waltham, Mass.*

UTILITY MILL: For precision milling of extrusions, sections and billets of aluminum or other nonferrous metals used in aircraft production. Has two 15-hp spindles and carriage feed speeds from 24 to 190 in. per minute. Length of bed, 60 ft; max work length, 55 ft; work space width and height, 24 and 17 in.; total width, 80 in.; spindle speed, 10,800 rpm with 180-cycle current. *Farnham Manufacturing Div., Wiesner-Rapp Co. Inc., Buffalo, N. Y.*

WIRE TAKE-UP: Completely automatic Class T dual wire take-up for use with dancer rolls taking wire from extruder or continuous vulcanizing machine. Take-up speed automatically synchronized with wire speed; reel speed automatically adjusted as wire fills reel to predetermined footage controlled by counter mechanism. When reel is full, wire is automatically transferred to empty reel at high speed by new traverse mechanism. Full reel automatically stopped by pneumatic brake. *James L. Entwistle Co., Pawtucket, R. I.*

PRESS: Model RS6½-14 press will bend mild steel plate up to 20 ft by ½ in. Can handle 14 ft sheets between end housings; has 6½-in. stroke and 18-in. throat depth. Overall dimensions, 20 ft long, 11 ft deep and 18½ ft high. Press bed extends 44 in. below floor level. Operated by electric air valve controlled by foot switch; speeds of 7 or 21 strokes per minute available through gear shifter. *The Cleveland Crane & Engineering Co., Wickliffe, O.*

BORING MILL: High-speed, single-spindle, horizontal boring, drilling and milling machine with 45 spindle speeds from 10 to 1300 rpm. Four-in. diameter nitralloy steel spindle has 30-in. travel. Machine includes hardened ways on bed and saddle, quick-acting, electrically operated positioning devices for table and headstock, swiveling pendent con-

trol, and simplified gear change requiring only two hand cranks. *Giddings & Lewis Machine Tool Co., Fond du Lac, Wis.*

Materials Handling

CHIP CONVEYOR: Removes metal chips, borings, and turnings from automatic or multispindle machines to eliminate need for shutdown for scrap removal. Formed steel hold-down meters scrap passing up belt to prevent jamming. Adjustable clutch slips if power load becomes excessive. *May-Fran Engineering Inc., Cleveland, O.*

HAND LIFT: Built into standard size hand truck to lift loads up to 500 lb as high as 54 in. from the ground. Hydraulic mechanism consists of lever-operated ram connected to lift pallet by chain and sprocket. Each stroke of lever lifts pallet about 1¼ in. Foot pedal controls lowering of load. *Allied Manufacturing & Sales Co., Chicago, Ill.*

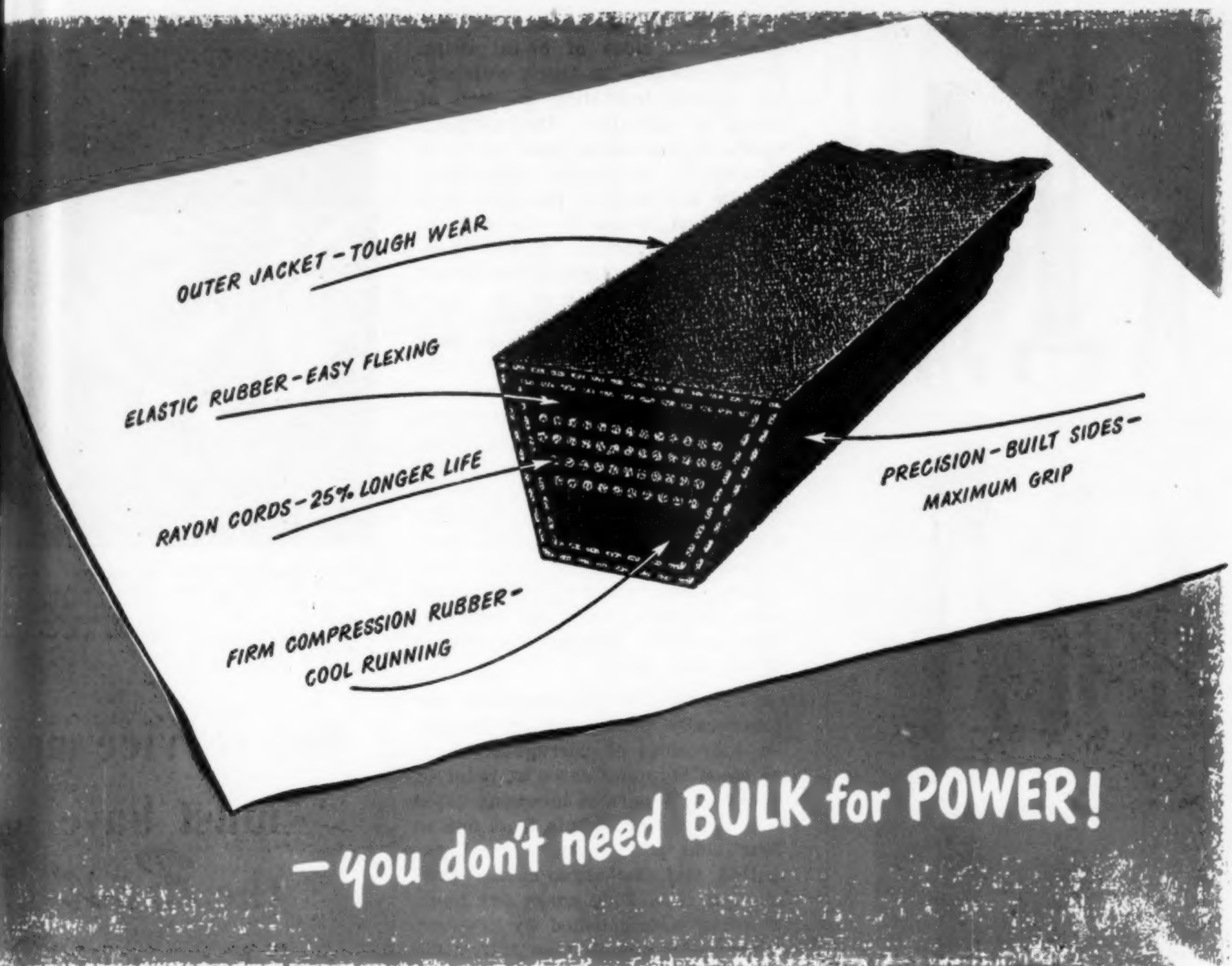
LIFT TRUCK: Capacity, 4000 lb. For stacking and tiering skids. With standard 24-in. platform in lengths from 30 to 48 in. Lowered height of platform, 7 or 11 in. With standard 83-in. mast, has 58-in. elevation. Right-angle stacks from aisles 24 in. wider than load length. *Lyon-Raymond Corp., Greene, N. Y.*

CHAIN HOIST: Type JC electric hoists in ¼, ½ and 1-ton capacities. For garages, industrial plants, shops, etc. Designed for hook suspension, with rope controls. Adaptable for trolley mounting. *Robbins & Myers Inc., Springfield, O.*

HIGH LIFT TRUCKS: Permit standard 83-in. overall height telescoping lift truck to tier loads in stacks over 16 ft high. Consists of extra set of front channels and additional lifting cylinder hung directly in front of regular telescopic channels. Device operated by separate controls. *Philadelphia Div., Yale & Towne Manufacturing Co., Philadelphia, Pa.*

TRUCK: Model 2500 truck for rapid materials handling. Has deck loading area of over 20 sq ft. Truck has one steering wheel, turning radius of 64 in. Powered by 13-hp, 2-cylinder, air-cooled Wisconsin gasoline engine, has three speeds forward and one reverse. Space beside driver's seat for long material. Designed for railroads, terminals, shops, industrial plants, etc. *Kalamazoo Manufacturing Co., Kalamazoo, Mich.*

ROLL HANDLING TRUCK: Special-purpose truck for handling 1500-lb rolls in storage room in complete darkness. Rolls measure 42 in. long, 32-



Gilmer Multiple V-Belts deliver power smoothly, efficiently, economically

Designed to fill the designer's needs! Gilmer Multiple V-Belts handle greater loads with the most compact constructions possible. Tensile section of strong, Rayon pulling cords—specially processed to minimize stretch—give these versatile belts a heart of pull! And all cords are locked parallel and cushioned all around to eliminate internal friction . . . rubber sections top and bottom each specially compounded, let belts run cool, smooth, and quiet even on short center and sharp-flex drives. Whatever your belt drive problems, it will pay you to investigate Gilmer V-Belts. Write for the Gilmer V-Belt Guide; it is complete with engineering data. Also we'll be glad to send you the Gilmer listing of special sizes in the Gilmer Mold List, form 135. Ask for it.

THE GILMER "TIMING BELT"* DRIVE —newest idea in power transmission!

A tooth-grip belt that makes positive engagement with axially grooved pulleys. Operates without slip or creep; speeds of driver and driven pulleys synchronize perfectly; angular velocity is constant. Complete information sent on request. *Reg. Trade Mark

Gilmer "Timing Belt" patented 1950

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Division of United States Rubber Company

Gilmer BELTS

Gilmer has the PULL!

in. diameter, are stacked 5-high along both sides of 54-in. aisles. Truck operates on track with signal system indicating position of truck to operator. Hydraulically operated positioning pads on truck prevent horizontal movement. Trucks are battery powered, with interlocked hydraulic controls to prevent malfunction. *Lewis-Shepard Products Inc., Watertown, Mass.*

Packaging Equipment

PACKAGE SEALER: Continuous hand rotary heat sealer designed for military packaging depots. For sealing of bags, barriers, innerliners, etc., made of heavier rigid materials such as kraft-backed foil, polyethylene, vinyl, or pliofilm. Makes 1-in. wide seal. Material fed into 8¾-in. long preheater section, then through spring-loaded ball-bearing rollers. Unit weighs 5 lb; two 150-watt elements can hold temperature of 600 F. *Pack-Rite Machines, Milwaukee, Wis.*

CASE PRINTER: New Markoprinter automatically prints display designs on four sides of corrugated, fibre or wood shipping cases at point of packing. Separates incoming cases for registration, prints from one to four side panels and top, if required, and discharges cases. Production rate, 2000 cases per hour. Printing accomplished by deep-cut rubber dies mounted on rotary die wheels. Built to individual size requirements, with wide size variation in each machine. *Adolph Gottscho Inc., Hillside, N. J.*

Plant Equipment

PLATFORM BEAM SCALE: Designed for accurate, low-cost industrial weighing where shock loading is present. All-steel scale employs roller-bearing mounted poises on main bar for rapid positioning. End loading platform gives same reading regardless of location of load on platform. Pit required, 11 in. deep. Capacities, to 6400 lb; platform sizes, 38 by 46 to 54 by 76 in. *The Yale & Towne Manufacturing Co., Philadelphia Div., Philadelphia, Pa.*

FORCED INDUCTION PUMP: For delivery of heavy lubricants and mastic materials which will not readily seek their own level. Hydraulically operated single air ram elevator used with heavy-duty air-motor operated drum pumps. Unit exerts 7110 lb pressure on material, will empty and clean sides of 400-lb or 55-gal drums. *Lincoln Engineering Co., St. Louis, Mo.*

ROD CUTTER: Guillotine 20E hydraulic cutter for cutting rod, bar shapes, chain, bolts, wire rope, cable, etc. Requires 2½ seconds for cutting

cycle. Head weighs 38 lb. Cuts rod and bars to $1\frac{1}{16}$-in. diameter, wire rope to $1\frac{1}{4}$-in. and cable to $3\frac{1}{2}$-in. Head connected to hydraulic pump with flexible hose. Powered by 2-hp electric motor or portable gasoline engine. Cutting blade has 60,000 lb thrust. *Manco Mfg. Co., Bradley, Ill.*

Testing and Inspection

SPECTROMETER: Direct-reading spectrometer for rapid analysis of metals including stainless steel and other similar materials. Provides pen-and-ink recorded analysis of samples, element by element, within two minutes. Can measure 25 elements, up to 20 simultaneously. *Applied Research Laboratories, Glendale, Calif.*

LEAK DETECTOR: Sensitive, inexpensive leak detector used with Freon or other halogen-containing gas. Model LD-01 detector employs simple circuit with three standard radio tubes housed in cabinet $10\frac{1}{4}$ by $7\frac{1}{2}$ by 8 in. Sensitive tube is 5 in. long with $1\frac{7}{16}$-in. diameter, has a 3-in. section of $\frac{7}{16}$-in. diameter. *Distillation Products Industries, Rochester, N. Y.*

GROOVE GAGE: Dial type gage for checking diameters of internal grooves and recesses, Truarc and O-rings, oil grooves, washer grooves, etc. Range setting made with knurled vernier nut adjustment. Settings made direct to gage block combinations or on micrometer. *Nilsson Gage Co., Poughkeepsie, N. Y.*

CONCRETE TESTING MACHINE: Capacity, 100,000 lb. For testing 2-in. cubes and 3 by 6-in. cylinders. Clear space between columns, $17\frac{5}{8}$ in.; max opening between ram and upper platen, $22\frac{1}{4}$ in. Ram travel is 3 in. at speeds to $1\frac{1}{4}$ in. per minute. Two 16-in. diameter dials for 0 to 100,000 lb and 0 to 10,000 lb graduated in 200 and 20-lb increments, respectively. Load measurement accurate to plus or minus 1 per cent or 0.2 per cent full scale capacity. *Baldwin-Lima-Hamilton Corp., Philadelphia, Pa.*

Woodworking Equipment

PORTABLE SAWS: For heavy construction jobs. Model 905 has 8-in. blade, weighs 19 lb, makes square cut of $2\frac{5}{8}$ in. or cuts through $2\frac{3}{16}$-in. lumber at 45 degrees. Has 115-v universal motor with no-load speed of 3600 rpm. Model 1000 saw cuts 1 to $3\frac{7}{16}$ in. at 90 degrees, $\frac{3}{4}$ to $2\frac{13}{16}$-in. at 45 degrees, has no-load speed of 3400 rpm. *Cummins Portable Tools, Div. of Cummins-Chicago Corp., Chicago, Ill.*

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